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Issued: June 1, 1977

(NASA-CR-152591) DEVELOPMENT OF A METHOD OF
ALIGNMENT BETWEEN VARIOUS SOLAR MAXIMUM
MISSION EXPERIMENTS Final Report (RCA
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FINAL ENGINEERING REPORT

Development of a method of alignment
between various SOLAR MAXIMUM
MISSION experiments.

Prepared for:
National Aeronautics and Space Administration
Goddard Space Flight Center
Greenbelt, Maryland

Contract NAS5-22347

RCA



RCA Government Systems Division
Astro-Electronics, Princeton, New Jersey

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1.0 INTRODUCTION

This report describes the results of an engineering study, performed by the RCA Astro Electronics Division, of the methods of alignment between the various experiments for the Solar Maximum Mission (SMM). The configuration studied consists of the instruments, mounts and Instrument Support Platform (ISP) located within the Experiment Module. Hardware design, fabrication methods and alignment techniques were studied with regard to optimizing the coalignment between the experiments and the Fine Sun Sensor (FSS). The proposed GSFC hardware design was reviewed with regard to loads, stress, thermal distortion, alignment error budgets, fabrication techniques, alignment techniques and producibility. Methods of achieving comparable alignment accuracies on previous projects were also reviewed. Amendments to the original scope of the study are embodied in Appendix A, Memoranda of understanding.

2.0 STUDY OBJECTIVES

2.1 Instrument Coalignment

- . Establishment of error budgets based on the alignment accuracies required for the various experiments and the FSS.
- . Review of coalignment data, prelaunch alignments and error budgets of previous projects.
- . Establishment of fabrication techniques to assure producibility and coalignment within the error budgets.

2.2 Mechanical Design

- . Review of the proposed GSFC platform and mount design with regard to interface requirements, loads, stress, stability, materials, tolerances, ease of assembly, producibility, compatibility with optical alignment techniques, and adequacy of methods of fastening and locking.
- . Review of existing static and dynamic loads analyses.
- . Review of alternate mount designs.

2.3 Thermal Design

- . Review of GSFC thermal design and analyses.
- . Analysis of the thermal distortion of the various instruments based on the fixed and flex mount configurations.

2.4 Optical Alignment

- . Development of an overall optical alignment plan.
- . Establishment of alignment datums for gravity effect measurements, coalignment of experiments to the FSS, and alignment maintenance verification at the system level.
- . Study of the use of templates in conjunction with optical alignment to meet the budgeted coalignment requirements.

3.0 STUDY DESCRIPTION AND RESULTS

3.1 Structural Analysis

During the course of the study the structural analysis effort divided chronologically into the following four major subdivisions:

- . Review of the MEGA structural analysis.
- . Stress analysis of the proposed mounts based on a static load of 2500 lb.
- . Preliminary dynamic analysis of the ISP and dummy instruments.
- . Stress analysis of mounts based on preliminary dynamic loads.

The review of the MEGA structural analysis is presented in Appendix B. As a result of this review, errors in the MEGA analysis were corrected and a static load of 2500 lb. was agreed to as the basis for the mount design and stress analysis. Load and stress analyses of the flex blade and flex mount assembly are presented in Appendices C through E. Flex blade geometry (length and thickness) has been optimized to provide an adequate margin of safety when loaded in the stiff direction as well as minimized moment transmitted when deflected in the compliant direction (see Flex Arm Design Chart at end of Appendix D). For the 2500 lb. load all margins of safety are positive including fasteners and dowel pins. Stresses are below the micro-yield for those items in the mount assemblies which would be alignment sensitive to permanent set.

Preliminary dynamic analysis of the ISP produced mount loads in excess of the 2500 lb. design load. These loads overstress the present fasteners in the mount assembly and would require significant design changes. However, this preliminary dynamic analysis shows

the need for more extensive analysis in two major areas. First, the stiffness of the planar grid adapter used in the preliminary dynamic analysis had a significant effect in reducing the ISP frequencies. In order to improve the design, a stiffer bolted interface is needed with the base support plane. At the time of the study, this interface was not well defined, and the initial estimates of the required adapter stiffness was shown to be inadequate by the preliminary dynamic analysis. Secondly, the analytical modeling technique used for Instruments 1 and 16 appears to underestimate the stiffness of these boxes and results in unrealistic dynamic response. The result of these two items is the prediction of mount loads far in excess of the initial 2500 lb. design load. Since these high loads are in doubt, an extensive dynamic analysis with realistic interface stiffness and instrument modeling is required prior to finalizing the flight mount design.

The modeling for the preliminary dynamic analysis is presented in Appendix D with descriptions of the ISP model and individual instrument models. The flex mount assembly stress analysis in Appendix E is presented in two parts: MA 75-12-4, dated 7 July 1975, is a detailed analysis for the 2500 lb. load condition, MA 7607-6, dated 4 June 1976, is an updated analysis for the worst case loading conditions based on the preliminary dynamic analysis.

3.2 Mount Design

3.2.1 Alternate Designs

Alternate designs were evaluated against the proposed three-point spherical mounting to the support plate via fixed and flex mounts. Various mount configurations were studied in terms of stability, thermal distortion budgets, ease of assembly, compatibility with optical alignment techniques and adequacy of

methods of fastening and locking. Two alternate designs (planar seat and intermediate plate) are presented in Appendix F, Mid-Point Review Vugraphs.

Planar seats were evaluated for their simplicity in both mount construction and experiment interface. These mounts, however, retain the traditional problems associated with alignment maintenance, namely, drilling for dowels following optical alignment and dowel extraction for instrument removal. These mounts do not lend themselves to the template alignments proposed for the instrument mounts.

An intermediate plate concept was evaluated as a simplification to the experiment interface. It provides for attaching mounts to the ISP on the normal grid pattern while allowing some latitude in the locations of the instrument mounting feet. Although this concept is not recommended for general use because of the weight penalty, it may provide a mounting solution for an instrument with mechanical interface problems.

3.2.2 Mounting Error Sources

The proposed NASA mount configuration was finalized with regard to alignment maintenance through disassembly and reassembly of an instrument on its mounts. Clearances between piloting parts were evaluated and recommendations made to provide for the ± 10 arc-second allowable mounting error budget.

3.2.3 Review of Mount Detail Drawings

GSFC detail drawings of the fixed and flex mounts were reviewed with regard to general design, dimensions, tolerances of fits, materials, finishes, fasteners and locking features. In the absence of assembly drawings TBS assembly drawings have been listed with relevant notes associated with the recommended design modifications.

The primary recommended design changes include a revised fixed mount (thermal similarity), revised thermal insulator keying (producibility), fabrication of the stub pilot and ball seat bore as a matched set (mounting error budget), and an increase in the size of internal mounting screws (stress). A complete listing of changes by GSFC drawing number is presented in Appendix G.

3.3 Thermal Analysis

3.3.1 Thermal Design

The designs of the fixed and flex mounts were reviewed thermally with regard to heat flow, temperature gradients and thermal distortion. The configurations were studied to determine the adequacy of the mount geometry, materials and finishes. Geometric similarity, polycarbafil insula or thickness, and bolt material were among the variables studied. The NASA fixed mount design was compared thermally with the proposed RCA fixed mount design which substitutes a titanium cruciform similar to the flex mount titanium blade in place of the aluminum cruciform. A comparison of these designs is presented on pages 16, 26 and 27 of Appendix F.

In addition the design of the platform was reviewed with regard to thermal stability.

3.3.2 Thermal Models

Thermal models of the mounts were developed for use in the computer aided thermal analyses. The NASA flex mount, NASA fixed mount and RCA fixed mount were modeled individually. These models along with their resultant orbital temperature gradients when used as FSS mounts are presented on pages 23 through 25 of Appendix F. The following effects were studied by computer analysis:

- . ISP Conductance Evaluation
- . ISP Orbital Evaluation
- . NASA Fixed Mount
- . RCA Fixed Mount
- . Flex Mount with Bolts
- . Flex Mount without Bolts
- . FSS Flex Pitch
- . FSS Flex Roll
- . FSS NASA Fixed Mount
- . FSS RCA Fixed Mount
- . Experiment 25 Flex Pitch
- . Experiment 25 Flex Roll
- . Experiment 25 NASA Fixed Mount
- . Experiment 25 RCA Fixed Mount
- . Experiment 29 Flex Pitch
- . Experiment 29 Flex Roll
- . Experiment 29 NASA Fixed Mount
- . Experiment 29 RCA Fixed Mount

3.3.3 Thermal Distortion

Angular deflections effecting experiment coalignment have been subdivided for analysis into distortion of the entire ISP and deflections produced by differential growth between the experiments and the ISP. Equations for bow in the platform have been developed based on differential growth produced by a temperature gradient through the platform. Equations for angular deflections of the experiments on the mounts have been developed considering deflections of the flex mount, as a guided cantilever, relating to the fixed mount. The equations and results of these analyses are presented on pages 19 through 22 of Appendix F. Platform distortion is presented as a function of coupling to the canister wall. Pitch and roll angular deflections of the experiments on their mounts are presented for experiments 1, 16, 29, 25 and 19 as well as for the FSS.

3.4 Coalignment Techniques

3.4.1 Optical Alignment Options

The following options for instrument coalignment have been considered:

- . Optical Alignment Using Lightweight Templates
- . Optical Alignment of Each Instrument
- . Gravity Compensation (Inversion Fixture)
- . Gravity Compensation (Load Cells)

Optical alignment by template is a technique which replaces each instrument with a lightweight template having an optical reference. The template for each instrument is used to position that instrument's set of mounts in the ISP prior to doweling the mount housings. This technique minimizes the gravity effects produced by instrument weight and allows optical alignment of the mounts to proceed prior to the receipt of instruments. An additional advantage is the reduced weight which must be positioned during alignment.

Optical alignment of each instrument involves securing of the instrument to its three mounts and positioning and securing the mounts within their cells to produce coalignment with the FSS. The instrument is then removed and the mount housings doweled to the ISP. To meet the coalignment accuracy requirements, gravity compensation must be provided for during this alignment. One technique utilizes an inversion fixture which allows data to be taken at both plus and minus 1 g with the instrument positioned for alignment with the data averages (zero g). A second technique provides load cells for each instrument to unload the ISP during optical alignment. This second technique requires a considerable fixturing effort associated with the optical alignment facility.

3.4.2 Coalignment by Template

The following two templating techniques were evaluated:

- . Master and Individual Templates
- . Dual Individual Templates

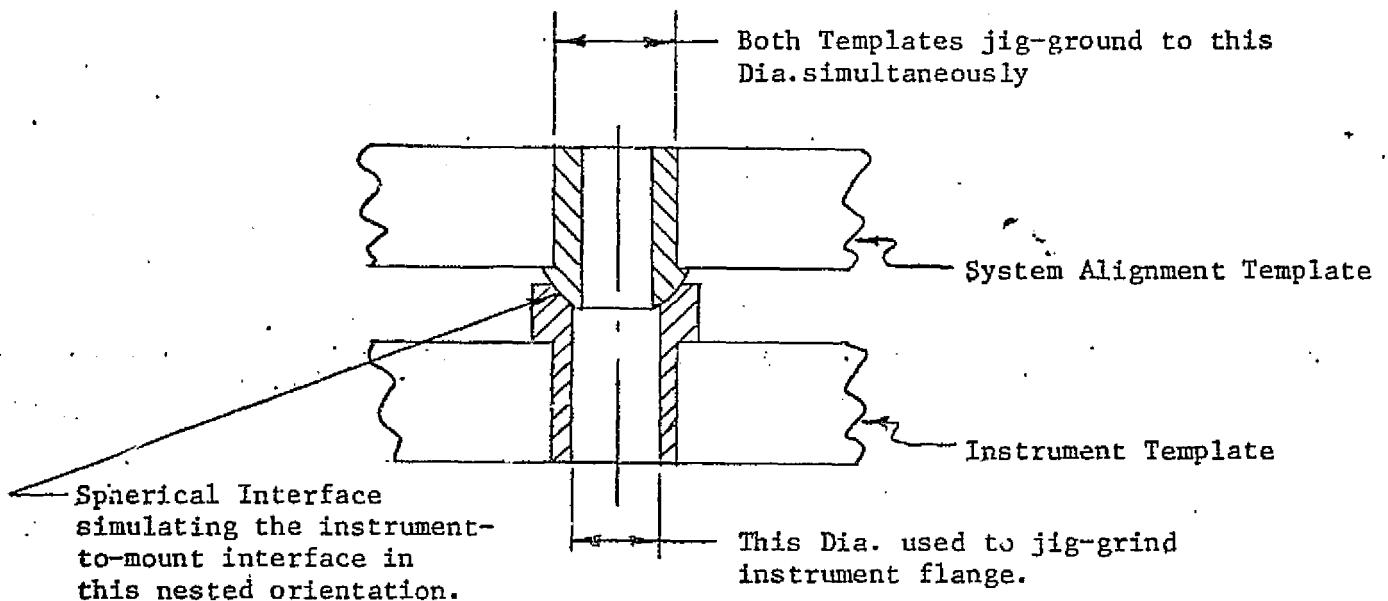
Using a master and individual template, the master is mounted to the ISP and the individual template mounted to the master and to the instrument mounts. The individual template is then used to locate the instrument mounts as well as drill the instrument flange. The following deficiencies are associated with the master template concept:

- . Spherical bearings in instrument flange must be installed within the ± 10 arc-second mounting budget.
- . Alignments including FSS will be to master template rather than FSS.
- . Cumulative tolerances in reference transfer between boresight, instrument mirror, individual template and master template.
- . Allowances and tolerances in bolted joint piece parts at the mounting points.

The following fabrication, assembly and alignment sequence is for the dual individual template concept of coalignment. Boresight alignment of each instrument to the primary reference (FSS boresight) is achieved with only measurement accuracy and spherical surface positioning accuracy as the residual errors:

- . Two blank individual templates are fabricated for each instrument (including the FSS) and temporarily doweled together for match machining.

- The three hole mounting pattern for the instrument is match machined (preferably jig-ground) in both templates simultaneously. This common diameter is sized to press fit inserts into each template.
- Precision inserts are pressed into each template. The system alignment template has inserts which simulate the instrument mounting flange geometry. The instrument template has inserts which simulate the mount geometry on the ISP.
- The templates are nested and clamped together simulating the instrument-to-mount interface at the three mounting locations. An optical cube is permanently mounted to each template. The residual misalignment between these cubes is recorded (templates nested and clamped).
- The instrument template is used to produce the mounting hole pattern in the instrument flange. The insert bore is sized to provide for setting up above each hole and jig grinding the instrument flange to size with some clearance to the template.



- . Inserts (spherical interface with mount) which become a permanent part of the instrument are pressed into the three mounting flange holes and the instrument mounted to the instrument template. Residual misalignment between the instrument boresight and the instrument template cube is recorded.

The alignment template for the instrument providing the primary reference (FSS) is used to fixture the three mounts for this instrument in the ISP. The mounts are doweled in position. This template cube is now the primary reference for all subsequent alignments.

- . The instrument templates in turn are used to fixture each set of three mounts in the ISP. The template is aligned and the mounts fastened by offsetting the alignment of this template's cube to the primary instrument's template's cube by the cumulative previously recorded residual misalignments (instrument template cube to alignment template cube, and instrument boresight to instrument template cube). This, in effect, provides alignment of the instrument boresight to the primary optical alignment reference. The mounts are doweled in position.
- . The instruments themselves are then mounted in place of the templates and alignments to their own cubes (not previously used in the alignment sequence) are measured in the 1 g optical alignment setup. These measurements become baseline for alignment maintenance through the environmental test sequence.

The tolerance studies for both the master and dual individual template concepts are presented in Appendix H, Template Alignment Tolerances. The master template concept produces a maximum error (RSS) of ± 51 arcsec. The dual individual templates produce a maximum error (RSS) of ± 7 arcsec which is within the ± 10 arcsec mounting error budget.

3.4.3 ATM Alignment

ATM experiment data, alignment methods and procedures were reviewed at Marshall Space Flight Center. The following highlights of that review are germane to the SMM alignment.

ATM optical alignment and alignment verification consisted of the following sequence:

- . FSS installation
- . Experiment installation
- . FSS and experiment alignment adjustment (counterbalanced)
- . Sequence of alignment verifications alternately free and counterbalanced.

The ATM instruments are in the 100-400 lb. class. Alignment adjustments vary with the instruments and include adjusting screws, adjusting mechanisms, parallel shims, tapered shims, and GIB plates. Following initial alignment, any required adjustments are made during the verification sequence in the counterbalanced condition. During the alignment and alignment verification sequence the loaded structure is allowed to settle eight hours after removal or application of load by the load cells. The FSS is mounted directly to the SPAR without alignment provision. Alignment of the FSS relative to the spacecraft is achieved by movement of the entire SPAR.

The ATM alignment summary indicated that alignment errors were generally under two arc-minutes but coalignment of the complete group within 30 arc-seconds did not appear likely.

A summary of the ATM investigation is presented on pages 43 through 45 of Appendix F.

3.4.4 Error Budget

The instrument coalignment error budget presented at the mid-point review is summarized on page 32 of Appendix F. Alignment errors were tabulated against the interface specification budget for the various error sources. Template alignment tolerances listed are for the master and individual template concept. For the dual individual template concept the mounting error is within the ± 10 arcsec budget. The "proposed alignment" listing is for direct alignment using the instruments themselves. The dual individual template concept had not been developed at the time of the mid-point review. At that time direct alignment of the instruments was proposed to meet the stringent mounting error budget.

Page 18 of Appendix F lists paragraphs in the SMM/Experiment interface specification which were deficient, at the time of the mid-point review, in specifying the accuracies required of the instrumentor with regard to mechanical mounting relative to the instrument boresight.

4.0 RECOMMENDATIONS

4.1 Design

The following recommendations revise the fixed and flex mount designs to provide configurations which meet the requirements of stress, producibility, alignment error budget, and alignment maintenance:

- . Change fixed mount configuration from aluminum cruciform to titanium cruciform in housing identical to flex mount - Alignment Error Budget.
- . Eliminate crossed keys in polycarbafil insulator and blade base (replace with dowels) - Producibility.
- . Fabricate stub pilot on flex and fixed members as matched sets with their respective ball seats (LAP bore) - Alignment Error Budget.
- . Increase size of internal mounting screws to 5/16 diameter - Stress.
- . Revise flex blade and cruciform thickness in conjunction with final stress analysis - Stress.
- . Eliminate close tolerance fit between polycarbafil insulator and housing bore - Producibility.
- . Provide witness mark keying (angular orientation) between lower ball seats and their respective fixed and flex members - Alignment Error Budget.
- . Provide dowels in mount assemblies between bottom plate and housing - Alignment Maintenance.

A number of the above recommendations have been incorporated in the revised GSFC drawings. Appendix G contains those recommended changes, by drawing, which were not yet incorporated as of 4 February 1976. Pictorial representations of the recommended configurations are on pages 5 and 16 of Appendix F.

4.2 Instrument Coalignment

The following recommendations are for a coalignment sequence utilizing lightweight templates in place of instruments to locate the fixed and flex mount assemblies in the ISP cells:

- . Instate the dual individual template concept including the fabrication, assembly and alignment sequence described in paragraph 3.4.2.
- . Align each set of mounts by rotating their individual template about the fixed mount. Flex mounts are positioned by moving the entire pre-shimmed housing in its cell in the ISP.
- . Shim during alignment at annular interface below lower ball seat. Grind or stone shims to size during the alignment sequence.
- . Torque mount housings to ISP during system optical alignment. Dowel housings and bottom plates to ISP following removal of templates.
- . Use removable positioners, as shown in Appendix F, pages 6 and 8, to rotate template and flex mount assemblies about the fixed mount.
- . When installing instruments, on pre-aligned mounts in ISP, support instrument weight externally (load cell for each instrument). Final torquing of any mount nut should be performed with all instruments supported externally.

- . Following instrument mounting and removal of load cells, alignment of the instrument cubes should be measured in the 1 g optical alignment setup. These measurements become baseline for subsequent alignment verification measurements.

4.3 Structural Analysis

In order to finalize the instrument mount design and prepare for the fixed base vibration test of the ISP, a final dynamic analysis should be performed. In generating the complete finite element model for this analysis, each instrument should be modeled separately with a modal analysis conducted for each unit. If the test hardware is available, a comparison should be made of at least the first mode frequency of each instrument on a fixed base with its analytical model. If discrepancies exist, the model should be changed accordingly. This insures adequate modeling of each box and the calculation of realistic dynamic loads in the mounting hardware.

The model used in the dynamic analysis should be based on the test configuration as closely as possible. This includes the interface adapter stiffness, instrument mounting stiffness, total weight and C.G. location, and box weight and inertia properties.

The detailed model described above should be used for the static, modal, and harmonic analyses of the ISP. This results in calculation of static and dynamic loads similar to the results of the preliminary analysis reported here. These loads should be used in a final detailed stress analysis of the IMP and instrument mounting hardware, thereby verifying the final design.

APPENDIX A

MEMORANDA OF UNDERSTANDING

Memorandum of Understanding

Minutes of Meeting held at NASA-Goddard on April 14, 1975

Subject: Research for Method of Alignment between Solar Maximum Mission Experiments

Attendees: D. McCarthy - NASA R. Packer - RCA
J. Pandelides - NASA W. Martin - RCA
A. Sherman - NASA (Part Time)

An initial meeting was held at Goddard on the above subject, the primary objectives of the meeting were as follows:

1. To establish a technical base from which to start the contract study awarded to RCA on the 17th April 1975.
2. To update RCA with the technical progress, program requirements achieved by NASA since the issue of the RFP.
3. To pass to RCA design information and analytical data performed by NASA to date, including alignment information from ATM and OAO.
4. To agree the Contract Work Statement and Phase relation to ensure details required for the NASA engineering model can be completed by the fourth week of the study.

1. NASA Experimental Instrument Mounting Platform

During the first quarter of 1975, NASA has designed and analyzed an Instrument Mounting Platform, using a configuration based on an experiment package selected at that time. The Experiment Module design interfaces with the Low Cost Modular Spacecraft (LCMS) via an aluminum truss system.

The experiment module consists of the instrument mounting platform supported via a three point mounting housed inside a lightweight structure thermally controlled, to which the experiment electronic packages are attached in the lower section.

The platform approx. 72" x 60" x 4" deep is an aluminum brazed system with 6061-T6 skins 0.125 gage, with an egg-crate core 0.062" gage, with suitable lightening holes. It supports eight experiments, total weight complement approx. 1111 lbs. and two Fine Sun Sensors weighing 11 lbs. each. This complement and placement is shown in Configuration 'A', which is the baseline RCA will use for the study. Each experiment is supported by 1

fixed, and 2 flex. mounts to be manufactured in titanium, and all experiments will be thermally isolated from the platform itself. The present plan is to use polycarbofil as the isolation material.

2. Mechanical and Thermal Test Model

a. Design and Fabrication

NASA will fabricate and test an instrument mount platform suitably mounted to an interface truss, together with a full complement of mass and thermally simulated dummy experiments.

The design is based on a stress and structural analysis performed by MEGA Analytical Research Services, who are also detailing the platform, platform supports and the instrument support. These drawings will be available to NASA and RCA in 2 weeks time. An early task is for RCA to design review these details, recommend to NASA the detail interface requirements, e.g. whether the 3 point interface between the experiment supports and the experiment should have a planer interface, or whether the mounts should contain spherical seats, this task to be completed by the end of the fourth week so that NASA can commence fabrication.

b. Detail Design of Instrument Platform

Because of the high load conditions imparted into the platform from the experiments, it is necessary to reinforce each pocket at the mounting hardpoints and carry the load through both skin face-sheets.

RCA will assess how stable such an assembly will remain under launch environment conditions.

c. Alignment of Mechanical Test Model

NASA plans to record the optical alignment position of each experiment to the Fine Sun Sensor, i.e. they will not bore-sight each experiment and shim to the correct axis alignment. They will measure only the stability and any shifts due to environmental testing. The final alignment technique will be developed during the RCA study.

3. Experiments

a. Experiment Placement

As mentioned earlier in these minutes, the weights, volumes and external dimensions and experiment placement that RCA will adopt for the study is as shown in Configuration 'A'. The co-alignment error budgets remain as detailed in the RFP and Work Statement.

b. Experiment Interface

A copy of the SMM/Experiment Interface Specification (provisional) was given to RCA. In general, it is intended that each experiment has an optical cube, to which the experiment itself is boresighted, the 3 point mounting interfaces will be machined to an accurate planar tolerance and will provide for a through bolt into the rigid and flex mounts.

c. Dummy Experiment Models

The mass simulated/thermal models are at present being detail designed. Copies of these drawings will be given to RCA in approximately two weeks for design review prior to fabrication at NASA.

d. Alignment-Experiments

NASA favors a master template-experiment template (one for each experiment) technique for controlling the interface for each experiment.

The master template, which has a master surface cube will:

- 1) Locate mounts in the instrument support plate.
- 2) Will be used to locate and machine holes in each experiment template

Experiment Template - Each experiment template will also house an optical cube which is used in conjunction with the cube on the experiment to ensure co-planar alignment between the flaying surfaces, the experiment cube and the boresight of the experiment itself.

RCA will develop the alignment technique and the optical test methods requirement with this requirement in mind.

4. Thermal Design

The instrument mounting platform is covered with an insulation blanket and the experiments are thermally isolated. The thermal gradients along the major axes are expected to be approximately 1°C . Through the IMP, i.e. face to face, the gradient is to be 0.05°C or less.

The thermal model of the IMP has 30 nodes for each face, and with 9 watts flowing from face to face, with a prediction of the 0.05°C delta.

The thermal model in total contains some 300 nodes to include the complete SMM Module, electronics and experiments. To date a steady state model of 45 nodes and 300 nodes have been run, and a transient run is about to be performed.

The electronic equipment at the base of the module is expected to operate at a temperature of $20^{\circ}\text{C} \pm 1^{\circ}$. While around the orbit the controlled thermal structure (IMP shroud) is expected to vary $\approx 2^{\circ}\text{C}$.

The inside of the shroud - sun facing panel has blanket insulation (painted black), the outside has a striped surface, black and white to achieve the correct thermal control.

The white paint selected is M.S. 74 silicate which is impervious to U.V. damage. The 310 n.m. orbit does not pass through a strong radiation belt and a stable thermal coating can be achieved.

The external walls of the thermal shroud have insulation blanket, except selected areas, which will act as radiators.

5. Work Statement

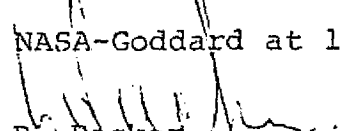
NASA confirmed the Work Statement tasks as outlined in the RFP and RCA Proposal. At the contract negotiation under task 2.1 the optimization of experiment placement was deleted. Task 2.4.1 Rastering Effects was also deleted.

RCA will start W.S. Task 2.1 - Initial review of all requirements, review all data recorded at Appendix of these minutes. RCA will also commence task 2.2.2 initial review of design, including details of the engineering test model as soon as the drawings are available from MEGA.

Several questions were outlined by NASA that should be answered.

- a. Does tightening of the flex mount bolts tend to move the experiments relative to the platform prior to pinning?
- b. Must determine the best sequence of instrument and experiment assembly - what effects does lg have on co-alignment?
- c. Should the templates weigh the same as the experiments, or be lightweight, so as not to load the mounts?
- d. Should the test be conducted unloaded? i.e. lightweight templates. (e.g. G.F.E.C. with hardened bushes)
- e. Should it be necessary to know the deflections and stresses in the instrument mounting plate due to each individual experiment as well as knowing and measuring the combined effects?

The next meeting is scheduled to be held at NASA-Goddard at 10:00 am on Thursday, May 8, 1975.


R. Packer
SMM Study Manager

APPENDIX

List of Documents and Memos received from NASA-Goddard.

<u>Documents</u>	<u>Dated</u>
- SMM Experiment Interface Spec. - NASA	3/18/75
- Program Validation for SMM-BBRC	F74-05
- Solar Maximum Mission Systems Definition Study (2 copies) - NASA	Nov. 1974
- A Stress & Structural Analysis Design Study for Solar Maximum Mission Instrument Support Plate. - MEGA	Feb. 1975

Memos

- Mounting of Instruments onto the Instrument Support Plate	3/25/75
- SMM Experiment Module Design/Test Guidelines and Assumptions	11/26/74
- Flight Load Analysis Task Assignments	3/5/75
- Table 6-1 Instrument Co-alignment for Pitch or Yaw	
- Table 6-2 Instrument Co-alignment for Roll	
- ATM Alignment Data - Draft Copy	5-25-71
- IMP Configuration A	
- Current Design of the SMM Experiment Module	4-7-75
- OAO Thermal Model	11/4/68
- Aerobee Star Tracker Test	12/7/70
- SMM Experiment Alignment	3/12/74
- Evaluation of SMM Plate and Flex Mount Test Samples	3/25/75
- Considerations for the Optical Bench of the Solar Maximum Mission Observatory	6/26/74

ATM

- Hydrogen - Alpha Telescope 1 Co-alignment Data
- ATM Maximum Alignment Error of Each Experiment to the Spar Based on a (1971) Tolerance Study
- ATM Instrument Alignment Errors as Measured
- Solar Instrument Boresight Error Measurement - BBRC 11/22/74
- Figure 4 Experiment Locations
- Figure 1 White Light Coronagraph
- Figure A-47 Rate Gyros, GSFC and HCO-A
- Figure A-48 FSS, NRLB and AS&E Experiment Mount Locations
- Figure A-49 Rate Gyros, Experiment Mount Location
- Figure A-50 HAO and NRL-A Experiment Mount Location
- ATM Basic Optical Alignment - NASA 1/1/71
- Pointing Position Relative to FSS
- In Orbit Long Term Alignment Stability - BBRC
- Ground to Orbit Alignment Shifts - BBRC
- Responses to Alignment Meeting Action Items - NASA 6/9/71
- ATM Alignment Modelling Study
- ATM Alignment Ad Hoc Working Group Meeting No. 3 - NASA 6/10/71
- HCO/ATM Instrument - Spar Alignment - BBRC 8/23/71
- SO55 Flight Instrument - Spar Alignment - BBRC 2/20/73
- HCO/ATM Instrument Spar Alignment
- HCO/ATM Instrument - Spar Alignment - BBRC 9/9/71
- HCO-NRL/ATM Co-alignment - BBRC 10/25/71
- HCO/ATM Instrument - Spar Alignment - BBRC 9/17/71
- HCO- NRL/ATM Alignment - BBRC 1/4/72
- Manufacturing Optical Alignment Procedure for ATM Experiments and Fine Sun Sensor (Flight Unit) - NASA 7/30/71
- ATM Alignment Ad Hoc Working Group Mtg. No. 3- Enc. 2-NASA 6/3/71
- ATM Alignment DATA - Draft Copy 5/25/71

A. Schnapf
May 13, 1975

75-RP-4
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Second 'Memo of Understanding'

Minutes of First One Day Conference held at NASA-Goddard on
May 8, 1975

Subject: Research for Method of Alignment between Solar
Maximum Mission Experiments

Attendees:	<u>NASA</u>	<u>RCA</u>
	D. McCarthy	R. Packer
	R. Federline	F. Gross
	B. Shisler	
	J. Diggins (Part Time)	

The first One Day Conference was held at Goddard on the above
subject, the objectives were as follows:

1. To relate to NASA the work completed to date on the study by
AED.
2. To obtain further technical information and an update from
NASA regarding their continuing work on SMM.
3. To view NASA's optical measurements facility in order to
continue task 2.2.1 and start task 2.4.3 of the Work Statement.

1. A) Fine Sun Sensor

AED has looked at the Flight history results from the ATM
mission, and it would appear that there was a general shift
in the co-alignment of all sensors relative to the F.S.S.
from which one can conclude:

- 1) The F.S.S. moved relative to all other sensors,
- 2) An error in the calibration and boresight of the F.S.S.
or the stimulus source had an error.

As the S.M.M. is flying a primary and redundant F.S. S.,
care must be taken to ensure that this error source is
carefully checked and the "Error knowledge" known.

NASA has already planned to mount both F.S.S.'s on a
common base, i.e., a single mounting system for both
instruments, and will ensure that the boresight to a single
cube is known. They will also ensure that the calibration
source is accurate.

B) Sample Test Plate

AED has completed a detailed study of the results of the mechanical test sample completed by NASA. It would appear that the residual stresses reported are in the noise level of strain gage accuracy; however, it would appear that some micro-yielding took place and that the results showed that 10,000 psi is the safe number for the micro-yield stress design case - which is close to the 12,000 psi that was recommended by AED for 2024-T4. NASA has been using 18,000 psi for 6061-T6. This is a very important design point and two action items resulted.

Action Item I

NASA will complete a hardness check to establish what the material was that was used - it is believed to be 6061-T6.

Action Item II.

AED will further research the Battelle literature for the micro-yield point of 6061-T4 through T6. AED will also investigate the literature for Titanium to establish whether 70,000 or 90,000 psi should be used in the design.

- C) AED reported their findings on a 1st cut run through of the 'Mega' dynamic and stress analysis. AED agrees in general terms with the modelling that has been done, but find it difficult to trace through the sequence, and exactly what was modelled. It is important that a loads matrix be developed, which will establish the effect of each sensor on the instrument plate, whether it is really necessary to use both skins to take the loads out in each case.

Three sizes of flex mount drawings have been provided by 'Mega', but again there is no definition of the size requirements for each experiment. The drawings also show a planar seat and a spherical seat, and a decision is required as to which method should be adopted.

Action Item III

AED will work with 'MEGA' for 1 day to gain further detailed knowledge of the modelling and attempt to develop the loads matrix, and from this matrix establish which flex mount sizes should be used. This meeting has now been arranged for Wednesday, 14th May. Dr. Sheffler (AED) and Mr. Honeycutt (NASA) will attend.

2. NASA outlined their continuing work; this included a first cut diagram of the sensors mounted on the IMP. NASA supplied copies of the first six dummy experiments and although the drawings do not detail the mounting interface to the experiments themselves, they do detail the layout of each experiment and the problems involved in designing a pick-up chassis for the flex mount interfaces, and the problems involved in mounting an optical cube.

NASA supplied unchecked drawings of the IMP and detail drawings of the attachments from the IMP to the platform support structure. This will allow AED to start task 2.2.2.

NASA supplied the results of the first steady state run of a 300 node thermal model, which will allow AED to continue the review of thermal effects under task 2.1.

AED reported that an initial look at the IMP thermal distortion during mission mode for approx. 0.1°F temperature gradient a 5 arc sec distortion can be maintained. This means that the steady state goal of 0.05° would appear to be ok but the transient conditions must be investigated. NASA is at present running a transient model and will give AED the results when the run is completed.

3. NASA Optical Alignment Facility

In a discussion with J. Diggins the present NASA thinking with regard to optical alignment of the instruments on the instrument support plate was reviewed along with the intended fixturing and the existing optical alignment facility. Error sources within the facility were discussed and stated to be equal to the ± 5 arc second error budget allocated for measurement instrumentation error in the SMM Experiment Interface Specification.

The Optical Alignment Facility consists of a theodolite mounted on a vertical tooling bar, a dihedral mirror used for azimuth reference, gantry-mounted relay mirrors, and a precision (± 1 sec) rotary table. The system to be aligned is mounted to the rotary table and viewed either directly or by means of relay mirrors above or below the test item.

The fixturing presently envisioned by NASA will support the SMM instrument support plate in a vertical orientation with the sun-viewing ends of the instruments facing up and viewed through the overhead relay mirrors. The second (roll) reference surface on each instrument will be viewed directly. The fixturing will have the capability of rotating the entire experiment package 180° about a horizontal axis to determine gravity effects. With the sun-viewing ends of the instruments facing down, relay mirrors placed below the inverted experiment package will be used to view the primary reference surface of each instrument. This fixturing is presently in the conceptual stage.

A. Schnapf
May 13, 1975

75-RP-4
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4. AED hours expended to date - as of May 2, 1975:

Task 2.1	Study of Thermal Effects and Stability of Instrument Platform	40 hrs.
Task 2.2.2	Review of documentation Review of 'Mega' Static & Dynamic Analysis	40 hrs.
Task 2.3.1	Initial review of Errors on A.T.M.	10 hrs.

Total 90 hours



R. Packer
SMM Study Manager

TROB/

F. Gross

Memorandum of Understanding

To A. Schnapf Location MS 57 Date June 9, 1975

From R. Packer Location MS 91 Telephone 2611

Subject Research for Method of Alignment between Solar Maximum Mission Experiments

Minutes of Second One Day Conference held at NASA-Goddard on June 6, 1975.

Attendees: NASA-GSFC RCA

D. McCarthy R. Packer
R. Federline F. Gross
B. Shisler
A. Sherman (Part Time)

The second 'One Day Conference' was held at Goddard on the above subject. The agenda for the meeting was as follows:

1. Instrument Flange Mounting Configuration
2. Flex Mount Size
3. Template Philosophy
4. Spherical Seat Mount
5. Thermal Deflection and Thermal Growth
6. Thermal Analysis - Transient - Thermal Model
7. Optical Alignment - effects of lg.
8. Methods used on other spacecraft - A.T.M.
9. Agenda and Date for Midpoint Review

1. Instrument Flange Mounting Configuration

RCA have been studying the details of the mount design proposed by GSFC and noted that three different instrument flange thicknesses are mentioned. The 'MEGA' report used 0.75", the drawings scaled 0.375", the ball details define 0.437". GSFC are designing the system for 0.50" flange thickness, and all drawings and analysis should use this thickness.

2. Flex Mount Size

Following the meeting held at Hightstown, whereby a nominal flex mount load of 2500 lbs. was adopted as a provisional working number to be used for sizing, RCA has completed a stress analysis

to size the blade thickness and shape. The analysis was based on: 2500 Load, 1.25 Factor of Safety, 70,000 psi microyield for Titanium. The analysis showed that the thickness required was 0.28", all other dimensions being the same as detail drawings supplied to RCA by GSFC. It was also pointed out that the 4-1/4" mounting bolts per mount were not large enough. It would appear that 5/16" bolts will probably be required. Further analysis will be required for final design.

3. Template Philosophy

RCA outlined a method that could be adopted for control of the interface between the IMP and Mount Systems related to control of the experiment mount flanges. The system uses a master template, with secondary templates, that register with the master template. The secondary templates would be furnished to each experimenter and would be used to control the holes in the instrument feet. The secondary template would also be used for installation of the rigid and flex mounts, to ensure their position and planarity - would pick up registers in the master template, therefore controlling the mount positions on the IMP. RCA stated that such a system could be designed to give a positional accuracy of 2 to 4 arc/minutes. It was recommended that other methods, such as shimming, vernier adjustment, etc., be considered for final alignments.

4. Spherical Seat Mount

RCA presented an error budget for the spherical seat mount, using reasonable manufacturing tolerances - in the order of 0.0005" for concentricities and ball to cup dimensional control. This would give an optical error of approximately 60 arc/secs. NASA expressed a viewpoint that Manufacturing tolerance control could be held to within say 0.0002" - which would mean an error in the order of say 20 arc/secs. This still leaves a problem of control of the 'Ball seats' into the experiment flanges relative to the 'Boresight' of the experiment - an interface that must be still worked. RCA will continue to study the design and improve it, if possible, and conceive a method of locking the interface. NASA expressed the thought that maybe it would be 'Cost effective' to go for more accurate machining tolerances, if alignment time, and tedious adjustments could be avoided.

RCA discussed planar mounts, using shim planes, with positive dowels. This would probably mean the use of reamers in the optical facility. NASA will check if reaming operations would be allowed in the facility. RCA will continue to study what errors the system could be controlled to, using such a design. It was also thought that some method of alignment adjustment may be required, such as micro-threads, for positional control, when such heavy experiments are involved.

5. Thermal Deflection and Thermal Growth

RCA presented results of thermal effects analysis due to:

- a) Assembly at room temperature of 22°C and mission steady state temperature of 18°C.
- b) Effects of a $\pm 1^\circ\text{C}$ change during an orbit period. These results show that due to thermal deflection the error is extremely small - both in Pitch and Roll, but effects due to the increase in length due to the difference in materials, etc., between the flex mount and fixed mount, is not small. For the F.S.S. and Experiment 1 the errors are 5.3 arc/secs in pitch and 8.03 arc/secs in roll. While these errors are well within the 15 arc/sec thermal error budget, the error could be reduced by making the fixed mount in titanium. NASA will alter the material to cancel out the error.

6. RCA are developing a thermal model of the mounting system and would have liked some transient thermal analysis data. NASA stated that the transient runs have not been run by them at this time, and it would probably be several weeks before results would be available. RCA will continue to use the 'Steady State' data.

7. Optical Alignment

RCA has not proceeded any further at this time with an optical procedure, but will refine the preliminary procedure during the next two weeks. RCA is attempting to come up with a viable method of cancelling the lg effect. Two solutions are possible:

- a) Light weight templates - if a system can be devised to guarantee positional placement when the experiment is interchanged for the template.
- b) The 180° vertical turnover in the optical facility, which is feasible but time consuming.

8. Other

RCA has not been able to perform the Work Statement tasks concerned with the review of A.T.M. detailed mount designs, because the information available is not in sufficient detail. Plans to visit M.S.F.C. will still be made, if we can establish that A.T.M. data is available.

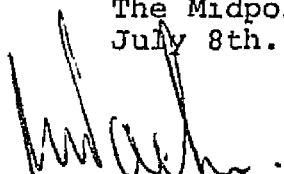
The technical officer requested that we use approximately 40 hours of the above task to analytically model an I.M.P. cell, to establish whether both skins carry the load in the present design.

A. Schnapf
June 9, 1975
Page 4.

9. Agenda for Midpoint Review

- a) RCA will present all the detail findings to date.
- b) Present details of spherical seat design and a planar seat design.
- c) Present the Thermal Model with latest inputs to reduce error to a minimum.
- d) Present a method of alignment.
- e) Give results of dynamics and statics of a single cell on the I.M.P.
- f) Present a method of testing.
- g) Present any information that can be obtained from M.S.F.C. on 'Flight' results of A.T.M.

The Midpoint Review will be held at NASA-GSFC on Tuesday, July 8th.



R. Packer
Manager, S.M.M. Study

cc: D. Brennan
J. Davey (4) (3 copies for NASA)
F. Gross
A. Manna
W. Martin
W. Metzger
R. Mercando

RCA

To A. Schnapf Location MS 57 Date July 14, 1975

From R. Packer Location MS 91 Telephone 2611

Subject Memorandum of Understanding

Solar Maximum Mission Study - Mid-Point Review
Minutes of meeting held on the above subject on July 8, 1975 at
NASA-GSFC.

Attendees - See Attachment I.

Agenda - See Attachment II.

1. A vu-graph presentation was made to NASA-GSFC in accordance with the contract requirements, at the mid-point of the study. The presentation followed the outline agenda as shown at Attachment II. Six copies of all vu-graphs were left with NASA, plus 3 copies of the stress analysis and 3 copies of RCA's F.E.M. of the I.S.P. Cell and Mount.
2. Four action items or questions resulted from the review to which RCA will respond. They are as follows:

Systems

Action Item I. Substantiate worst case alignment errors, due to mechanical tolerances, that could be achieved, if a template only alignment procedure was to be adopted.

Thermal

Action Item II. Check calculations on error magnitude of the Fine Sun Sensor in yaw.

Action Item III. Check the calculations and assumptions that resulted in RCA's prediction that the error magnitude due to temperature changes while 'On orbit' could be as much as 3.5 arc/secs in any 5 minutes of time.

Stress

Action Item IV. Check the 'Margin of Safety' in the face sheets of the I.S.P. which was stated to be 80% - a quick check shows that this case was for 'screw tear out' only and the overall margin is probably in the order of 20%. RCA will confirm this margin.

A. Schnapf
July 14, 1975
Page 2.

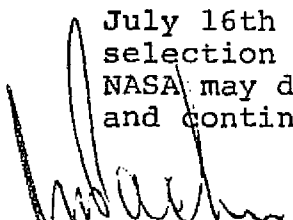
Discussion

At the end of the presentation, a discussion was held which centered on three areas.

1. RCA were asked to look at the introduction of a 3rd Flex mount angled at say 45° (dependent on experiment geometry) to replace the fixed mount - a claim being made that this would minimize stress levels into the experiments themselves. RCA feels that such a system could induce other major problems such as dynamics stability, lower first modes, be difficult to handle in final alignment. RCA will take a more in-depth look at such a system prior to submitting final conclusions to NASA.
2. Some difficulty is being experienced in defining the actual loads expected into the I.S.P. from each experiment, and therefore the loads that the 'Fixed' and 'Flex mount,' will see. This is primarily due to the fact that 'Mega Research Analysis' used a different (earlier) experiment configuration, to the present Configuration 'A' with the 'fixed' and 'flex' mounts also in different positions.

RCA were asked to quote the number of labor hours and computer time required to rework the analysis at AED to obtain a firm design analysis for the loads to which the flex and fixed mounts should be designed. This quote will be given to NASA formally during the week ending 18th July '75.

3. Because of the amount of detailed information presented to NASA, the technical officer stated that NASA would require several days to consider all the recommendations made, pinpoint and select a single mechanical design for the experiment mounting system, which would allow RCA to proceed into the final phase of the study. The writer agreed to visit GSFC on Wednesday, July 16th to discuss the tasks remaining, receive NASA's design selection and direction, and assess whether all the tasks that NASA may desire are within the present 'Scope of the Contract' and continue with the study accordingly.



R. Packer
SMM Study Manager

eb

cc: D. Brennan
J. Davey (11 copies)
F. Gross
W. Martin
A. Sheffler

ATTACHMENT I.

ATTENDEES

Phone No.

NASA

J. L. Diggins	982-4131
J. Donley	-6260
R. Federline	-5621
P. Honeycutt	-4375
R. Leland	-4696
L. Linstrom	-5628
J. Mason	-6477
D. McCarthy	-6010
J. Pandelides	-6952
A. Sherman	-5405
B. Shiffler	-4380
J. Stivaletti	-4332
L. Veillette	-4908
M. W. Wilson	-4181

RCA

448-3400

R. Packer-	Study Manager	-2611
D. Brennan-	Marketing	-2713
F. Gross-	Mechanical Design	-3215
W. Martin-	Systems and Thermal Analysis	-2494
A. Sheffler-	Mechanical Analysis	-2952

ATTACHMENT II.
SOLAR MAXIMUM MISSION STUDY
MID POINT REVIEW
8 JULY 1975

AGENDA

	<u>Presenter</u>
1. Introduction	R. Packer
2. Objectives of Study	R. Packer
3. Review of NASA Flex Mount Design	F. Gross
. Recommended Design Changes	
. Scheme for Alignment Adjustment	
. Method of Alignment and Features for Spherical Seat Mount.	
4. Review of a Planar Mount Concept	F. Gross
5. Intermediate Plate Concept	F. Gross
6. Alignment by Use of Templates	F. Gross
7. Fabrication, Assembly and Alignment Sequence.	F. Gross
8. Fixed Mount Design	F. Gross
9. SMM/Experiment Interface Specification	F. Gross
10. Thermal Design Overview	W. Martin
. ISP Thermal Stability	
. Thermal Model of Flex and Fixed Mount	
. Orbital Tolerance Budget of NASA Design with RCA Proposed Changes.	
11. Optical Alignment Plan	W. Martin
12. Systems Composite Error Budget	W. Martin
13. Review of MEGA Analysis	A. Sheffler
14. Model of ETM ISP Cell and Flex Mount Design.	A. Sheffler

AGENDA (Cont.)

	<u>Presenter</u>
15. Stress Analysis of Proposed Design	A. Sheffler
16. ATM Investigation	R. Packer
. Basic Design Criteria	
. Alignment Method	
. Alignment Procedure	
17. Environmental Testing	R. Packer
18. Task Phasing Plan	R. Packer
. Requirements to Complete Study	
19. Discussion	

IRCA

MA 75-12-6

To R. Packer Location MS 87 Date 25 July 1975

From A. Sheffler Location MS 105 Telephone 2494

Subject Review of Solar Max Mission Analysis and Technical Proposal

Mr. Steve Brodeur of NASA/GSFC reviewed the analysis done to date on the ISP reflex mounts and the proposed analysis of the entire Instrument Support Plate (ISP). There was a general understanding and agreement reached on all issues. The present analysis was accepted as correct, and the proposed analysis techniques were considered worthwhile improvements over the MEGA analysis. The following topics were discussed.

Stresses in Flex Arm

A basic agreement was reached regarding the flex arm stress analysis method, although different analysis methods gave slightly different results. It was not considered a serious problem, but some further investigation will be carried out in an attempt to solve this puzzling "loose end".

Discussion of FEM of the Flex Arm Assembly

Modeling techniques were discussed including equivalent stiffness calculations, load paths, and the relative accuracy of the stress levels. We were in agreement that the model was accurate as is, and the resulting analysis was acceptable for supporting the design activity.

Discussion of Proposed ISP Analysis

• Description of FEM:

Modeling techniques were discussed regarding the STARDYNE plate elements to be used in the proposed ISP analysis. The proposed method was accepted as being an improvement over MEGA's approach. Discussion of the modeling techniques used for the ISP support structure and the reinforced areas on the ISP led to the decision that additional plate elements would be required, but these would be a negligible increase in the cost and effort.

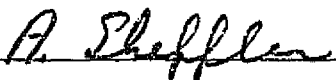
• Applied Loads:

It was agreed that the static loads are well defined and the analysis should be minimized to the critical cases wherever possible. The dynamic load cases, however, must be considered in more detail. Rather than estimating the dynamic input, it was suggested using a stick model analysis to determine realistic inputs to the ISP. George Honeycutt of GSFC is presently developing a stick model to determine transient flight loads. I explained that our present proposal assumed the dynamic input would be defined by mutual agreement between NASA/GSFC and AED. If the Honeycutt analysis were available, the loads could easily be defined. On the other hand, the loads could be calculated by AED using the Honeycutt stick model with the DELTA base input. Since our proposal did not include the dynamic loads analysis, it must be treated as an additional item. It was agreed this is a reasonable approach.

• Instrument Box Stiffness:

Since the engineering test unit (ETU) will contain simulated instrument boxes, the analysis should model the same inertia and stiffness as the test items. It was agreed the proposed modal analysis for each box as a subsystem was not required. Only critical boxes will be modeled in this manner.

In general, the proposed analysis was enthusiastically accepted, and a good understanding was reached by both parties.



A. Sheffler

jah

dc F. Gross
W. Metzger

NEGRAD

To A. Schnapf Location MS 57 Date 6 August 1975

From R. Packer Location MS 67 Telephone X2378

Subject Research for Method of Alignment between Solar Maximum Mission Experiments. Memorandum of Understanding

Minutes of Third One Day Conference Held at NASA/GSFC on the 31st July 1975

Attendees:	NASA	RCA
	D. McCarthy	R. Packer
	R. Federline	F. Gross
	B. Shiffler	D. Brennan
	S. Brodeur	

1. Alignment

RCA presented an alignment plan technique using matched templates for each experiment, together with a tolerance matrix and resultant mechanical errors. Such a system will meet the NASA error budget specified and establishes a unique method of generating the 0 to 1g changes, establishes a known test datum and allows the placement of the experiments, that have a given and/or measured 'boresight' error knowledge, such that each experiment can be accurately aligned to the F.S.S. This method does not burden the experimentors with tight tolerances, and was adopted as the method to be implemented.

2. Detail Design of Flex and Fixed Mounts

RCA and NASA have agreed on the detail design and type of mounts, but the final sizing cannot be finalized until the final loading conditions can be established. It was decided to delay the final detail design until the 'load profile' is established.

3. Mechanical Analysis & Loads

NASA has decided to award RCA a change in scope - to complete a computer analysis of the IMP plus experiments, to determine the loads, in order to finalize the design of the mounts. A work statement of requirements has been given to the NASA contracts office for transmission to RCA. This work statement will be costed by RCA and returned to formalize the change in scope. The ROM cost given to the Technical Officer was in the order of 15,000.00 which represents RCA's envisioned tasks required, which has also been reviewed by S. Brodeur as being the task outline necessary.

6 August 1975

4. Test Plan

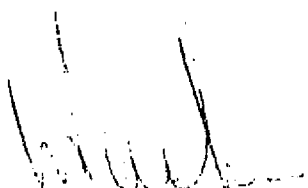
RCA is now in a position to outline a test plan that will satisfy all the testing requirements and will start this work during early August.

5. Final Study Report

RCA is now in a position to commence the draft of the final report. It will not be finalized until the Mechanical Analysis has been completed, loads established and final detail drawings of the experiment mounts completed. NASA stated that the delay in completing this extra work before establishing a final design is important and will not impact the fabrication date for their engineering model.

6. Assembly of the Engineering/Thermal Model by RCA

RCA presented the Technical Officer with a ROM of \$39,000.00 for the assembly of the engineering model as requested at the previous meeting. This quote will be given to the S.M.M. project manager for consideration. It was recommended by RCA that this additional work could also be added as a 'change in scope' to the present contract, and D. Brennan (RCA) will discuss the possibility with Messrs. Burr and Pandelides.



R. Packer
S.M.M. Study Manager

cfm

cc: D. Brennan
J. Davey (10 copies for NASA)
F. Gross
A. Manna
W. Martin
R. Mercado
W. Metzger

11/15/75

MA 75-12-7

To R. Packer Location MS 67 Date 17 November 1975

From A. Sheffler Location MS 77 Telephone 2610

Subject Solar Maximum Mission Technical Meeting: Memorandum of Understanding

A meeting was held at NASA/GSFC on 14 November 1975 to establish the information required for RCA to perform the analysis of the Solar Max Mission panel and the design of the instrument fixed and flex mounts. The following persons attended the meeting:

S. Brodeur--GSFC
R. Federline--GSFC
G. Honeycutt--GSFC
A. Sheffler--RCA

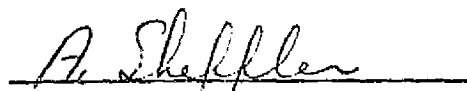
Minutes of Meeting

1. Drawings of the SMM ISP substructure and experiments to be used in the vibration and all design details were given to Al Sheffler and described by Bob Federline. This information will be used by RCA to model the SMM structure and ETU experiments. RCA will review and contact Steve Brodeur for any further clarification or information.
2. Discussed notching criteria and all parties agreed that, lacking a flight loads analysis, the only reasonable criterion is to notch based on the bending moment (loads in MMS/Delta adapter) at the vehicle interface under quasi-static loads of 17.8g longitudinal and 3.3g lateral. RCA will run a harmonic analysis of their model using a 1g flat input and scale the response to achieve the same acceleration response pattern in the ISP as obtained in the Honeycutt/Brodeur MMS/SMM model.
3. Information required to define the foregoing ISP acceleration response pattern will be supplied by Honeycutt/Brodeur to RCA in the time period:

26 November	Earliest date
5 December	Latest date

This information will consist of:

- a. Plots of acceleration vs. frequency for 10 points in the ISP: 3 along the tip, mid and bottom and at the side strut attach point.
 - b. Printouts of same information for all other points in ISP and strut attachment points.
 - c. Off-axis responses in printout form for above points.
 - d. Mode shapes and frequencies of GSFC SMM/MMS model.
4. General agreement was reached on GSFC information:
 - a. 15-20 modes to be used in model formulation.
 - b. Excitation to 50-75 Hz range.
 - c. 5 percent structural damping for lower modes but 2 to 3 percent on plate modes.
 5. The possibility exists that, even with input to the RCA SMM scaled to give the same response as seen when the GSFC model is limited to the static bending moment, the loads in the ISP may still exceed allowable design loads and may require further notching. This would occur most likely in ISP plate modes at approximately 18-20 Hz. Lacking a flight loads analysis, the consensus was to wait and see if this is really a problem.
 6. Static load levels to be used by RCA as part of their stress analysis of the ISP will be supplied by GSFC. These static loads constitute one part of the stress analysis; the other part will be based on the dynamic loads determined in Task C of the RCA Proposal. The larger loads and stresses will determine the design and/or margins of safety. RCA will use these loads to design the fixed and flex mounts.
 7. Quasi-static design loads are 17.8g longitudinal and 3.3g lateral. These loads provide for a 10 percent margin. No margin of safety should be included in the stress margin calculations. RCA will strive for a zero margin design in the fixed and flex mounts based on a microyield criterion.


A. Sheffler

dc F. Gross
W. Martin
W. Metzger

jal

MA 76-7-1

To R. Packer Location MS 67 Date 21 January 1976

From A. Sheffler Location MS 77 Telephone 2610

Subject Solar Max Mission Loads Analysis--NASA Status Review

A status review of the SMM Loads Analysis was held at NASA/GSFC on 14 January 1976 with the following persons attending:

F. Gross--AED	R. Federline--NASA
A. Sheffler--AED	S. Brodeur--NASA

A summary of the analysis done to date was presented according to the attached agenda. The following topics were discussed in some detail:

1. The stiffness of Instrument Boxes No. 1 and 16 are inadequate as presently shown. There was general agreement that a re-design was expected could be done easily.
2. The total mass and C.G. location is acceptable, but could not be verified. NASA has no current estimate of this information.
3. The major panel resonant frequencies are 10.3 Hz and 16. Hz. One "soft" member in the structure is the support beam at the fixed ball support. This beam allows considerable in-plane motion of the panel along the Z-direction. It was agreed that an additional static computer run should be made to estimate what additional stiffness is required. In particular, this analysis should determine the effects of extending the .090 inch doubler to the full length of this support beam.
(NOTE: The pedistal support stiffness was also found to be inadequate. This was reported to NASA by telecon 1/20/76.)
4. The notching criterion as defined in the proposal was reviewed and accepted as valid for this analysis.
5. It was agreed that no factor of safety should be used in the flex arm stress analysis. However, since the dynamic test levels will be to full qualification loads, the margin of safety should be maintained at 25 percent for any dynamic load conditions.
6. For thermal reasons, titanium screws will be used to attach the flex arm to its housing. This should be noted for the stress analysis.

7. An interference fit was detected in the ball seat assembly. NASA will review the drawing dimensions and correct as needed.
8. A January 30 completion date for the flex arm analysis is acceptable to meet the NASA schedule.

A. Sheffler

A. Sheffler

jal

dc F. Gross

F. Hayes

W. Metzger

G. Varadarajan

ADGENDA

SOLAR MAXIMUM MISSION STATUS REVIEW

- State of Problem
- Description of FEM
 - . Plots
 - . Weight comparison
 - . Equivalent face sheet stiffness
 - . Equivalent core stiffness
 - . Analyses
 - static
 - modal
 - harmonic
- Results of FEM Analysis
 - . Modal
 - 10.3 Hz, 16. Hz, and 39. Hz panel modes
 - box modes and box stiffness
 - . Harmonic
 - peak response frequencies (lg base input)
 - notch levels
 - nominal base input
 - flex arm loads
 - . Static
 - flex arm loads

ADGENDA (cont.)

• Flex Arm Analysis

- . Load summary
- . Description of FEM and applied loads
- . Allowables
- . Margins of safety
- . Recommendations

APPENDIX B

REVIEW OF MEGA STRUCTURAL ANALYSIS

Internal Correspondence

MA 75-12-2

To R. Packer Location MS 87 Date 20 May 1975

From A. Sheffler ✓ Location MS 105 Telephone 2494

Subject Review of Structural Analysis of Solar Maximum Mission Instrument Support Platform

On May 14, 1975, I attended a review of the analysis conducted by MEGA Analytical Research Services of the SMM Instrument Support Platform (ISP). The following persons were present at the meeting:

Dr. Richard Dame--MEGA
Hank Cornille--MEGA

George Honeycutt--NASA (Part Time)
Bob Fedorline--NASA (Part Time)
Bruce Shiffler--NASA (Part Time)

The agenda for the meeting is attached as an appendix. All topics were discussed in sufficient detail to evaluate the analytical methods. This discussion yielded several items which affect the ISP design:

1. Design Loads

The loads used in sizing all members are based on the static loads in the Delta S-320-G and the Shuttle JSC 07700. MEGA indicated that no dynamic environment was specified. In order to obtain some assessment of the dynamic response loads, MEGA assumed the highest lateral harmonic input at the base of the ISP supporting structure to be equal to the largest static lateral loads (4.56g). It was assumed that this dynamic input was conservative since the Delta spec is only 1.5g lateral input. However, this assumption is inconsistent and could lead to an unconservative design for the following reason: The Delta spec defines a 1.5g lateral input at the base of the spacecraft. In the present design, the base of the spacecraft is not at the base of the ISP, but, rather, it is located below the three support modules. The dynamic response of this stackup should be used to determine the proper input to the ISP, and this could be greater than 4.56g. A similar dynamic analysis should be conducted for the shuttle configuration to determine realistic

dynamic loads in the ISP. At present, this type of analysis has not been done; and no indication was given that it would be in the near future. Therefore, the results of the dynamic analysis are in question.

2. Margins of Safety

MEGA indicated that all margins of safety for the ISP and instrument mounts are based on the microyield stress allowables with no factor of safety. It is MEGA's belief that the design qual static loads already contain substantial factors of safety so that it would be unrealistic to compound these factors. This does not comply with the AED practice of using 1.25 ultimate factor of safety when the item is tested to full qual level loads. For example, under the AED criterion, the loads for the Delta are 18g qual test and 22.5g design ultimate. Microyield is taken as a design ultimate condition because it determines the success of the mission.

3. ISP Frequency Requirements

MEGA stated that no minimum frequency was specified for the ISP. However, it is apparent that the first resonance should avoid the 15-21 Hz POGO region. The present design shows a hard-mounted frequency of 19.6 Hz. MEGA is attempting to raise this frequency to get above the POGO region. However, this could be troublesome since the resonant frequency will drop when the ISP is mounted on the service modules. Therefore, it may be beneficial to design for a coupled response at less than 15 Hz.

4. Detail Drawings

Apparently, MEGA produces preliminary drawings which are completed at GSFC. Only a few drawings were available at this meeting. Detail drawings of the graphite/epoxy support strut and revised drawings of the ISP support brackets will be supplied by GSFC.

5. NASTRAN Modeling Techniques of ISP

The modeling of the ISP has sufficient detail to be used for a realistic stress analysis. Furthermore, MEGA's procedure of using minimum section properties in the stress model does lead to the maximum panel stresses. The use of average section properties in the dynamic model is also an acceptable approach for a realistic modal analysis.

One possible discrepancy, however, exists in the shear coefficients of the PBAR 51 elements used in the ISP model. The K1 and K2 shear factors are apparently reversed. This would increase the transverse shear stiffness of the ISP and reduce its in-plane stiffness. This discrepancy could easily be checked by MEGA.

6. Modeling of Instrument Boxes

The instruments located on the ISP are modeled as rectangular boxes with six quad-plate elements forming each box. The weight of each box is equally divided into eight mass points at the corners. This adequately represents the mass distribution by accounting for the total mass and the rotational inertias of each box.

The stiffness of the boxes is represented by assuming a 0.125 in. thickness for the quad-plates forming each box. Since no box stiffness is specified, this representation is assumed to be typical. MEGA has not looked at any other box stiffness with similar geometry. However, it is apparent that each box forms a redundant load path with the ISP and a realistic box stiffness should be defined.

As a quick check of the box stiffness used by MEGA, a 190 lb box (12" x 19" x 44") was selected. Assuming simply supported ends across the 44 in. length, the box resonant frequency is approximately 500 Hz. This ignores the torsional compliance which would result from the three point support, but it does indicate a high box stiffness in the present analysis. Since this same box produces the highest loads in the flex mounts (5800 lb), the effects of box stiffness should be reviewed. At the meeting, MEGA indicated the box stiffness effects could not be adequately discussed because only one box configuration has been analyzed.

7. Flex Mount Analysis

MEGA has taken the fixed mount for each box to be located on the "lower-inboard" corner of each box. The two flex mounts are then placed at the opposing corners to allow for thermal growth. In the ISP model, the flex mounts are included as equivalent beams having the stiffness of the flex arms. The canister which attaches the flex arm to the ISP is also modeled as a set of equivalent beams. These properties, however, do not account for the stiffness of the bolted joints or for the difference in length between the equivalent beams and the canister. The result is that the flex joint in the model is more flexible than the actual structure. This could show

lower loads in the flex arms than actually occur. This is not an easy question to answer without further NASTRAN analysis because of the coupling between the panel, the flex mounts, and the box.

8. Flex Mount Loads

MEGA has produced three flex mount designs with the following load capacity:

Type	Allowable Load in Stiff Flex Arm Axis
Large	5800. lb
Medium	2370.
Small	980.

All parts of the flex arms are sized for near zero margins at the allowable load except for the higher margins on the polycarbofil inserts. Since only one support configuration was considered by MEGA, no information is available to determine the effects of moving a box C.G. with respect to the mount locations. This would require further analysis along with a reliable box stiffness definition.

9. ISP Support Structure

The panel support structure consists of brazed members with a minimum of joints. The NASTRAN model appears to properly represent this structure and accounts for the pinned and ball joints. The support structure is sized for the conventional yield criterion (not micro-yield).

10. Launch Loads Analysis

The NASTRAN model of the ISP and support structure will be reduced to approximately eight degrees of freedom for use by the launch vehicle contractor to determine flight loads. This reduction in the complexity of the model will be done by NASA. The resulting transient flight loads may be applied to the detail model, but at this time, no commitment has been made.

APPENDIX

Agenda

Solar Max Mission--Instrument Support Plate

- Review Design Constraints
 - . Loads environment (static and dynamic)
 - . Weight and geometry restrictions
 - . Frequency requirements
- Review Assembly Drawings and Material Selection
- Scan Drawings in Increasing Order of Detail
- Review NASTRAN Modeling Techniques
 - . Basic geometry and detail
 - . Element properties
 - . Static analysis: load cases
 - . Modal analysis: mass distribution
 - . Harmonic analysis: loads and damping
- Review Procedures Used to Size Members
 - . Core, core walls, faces, flex mounts, etc.
- Description of Instrument Complements Considered
 - . Effects of moving boxes with respect to ISP
 - . Loads in panel
 - . Loads in flex mounts

- Box Mount Loads
 - . Effects of box C.G. with respect to fixed and flex mounts
 - . Load paths into panel core and faces
 - . Effects of box location with respect to ISP on mount loads
 - . Effects of box orientation on mount design
- Define Required Box Stiffness--If Any
- Design/Analysis Procedure for ISP Support Beams
 - . Joint details
 - . Materials selection
 - . Loads summary
- Launch Loads Analysis
 - . Description of dynamic model to be sent to launch vehicle contractor
 - . Use of resulting launch loads

A. Sheffler

A. Sheffler

jah

dc F. Gross
W. Metzger

APPENDIX C

STRUCTURAL ANALYSIS OF SMM FLEX ARM

HUGAN

MA 75-12-3

o R. Packer Location MS 87 Date 19 June 1975

From F. Hayes Location MS 105 Telephone 2985
A. Sheffler MS 105 2494

Subject Structural Analysis of Solar Max Mission Flex Arm Assembly

This report presents the stiffness and stress analysis of the flex mount for the Solar Maximum Mission Instrument Support Panel (ISP). A STARDYNE finite element model was constructed for a typical flex arm assembly including the corner support fittings (Figures 22-24) for the flex mount housing. Member loads, stresses, and deflections were obtained for unit applied loads at the flex arm/instrument interface (see Figure 1).

The STARDYNE finite element model is shown in Figures 1 through 5. The housing for the flex mount is modeled as an octagonal network of beams as shown in Figure 1. This beam model takes into account the polycarbofil insulator (in element 49) as well as load transfer in the integral top plate (Figure 2) and the attached bottom plate (Figure 3). The corner fittings used to mount the flex arm assembly are modeled as beam elements screwed to the panel face sheets and to the core webs. These screw locations are node points in Figures 5, 22, 23, and 24. The beam elements then define the loads in the screws and the required preloads for sufficient friction to eliminate slipping.

Figure 6 shows the restraint system while Figures 7 and 8 give material constants and beam section properties. Figures 9 and 10 show bending moment and stress distribution in the flex arm.

The attached figures contain the following information which is presented here for use in the detail design:

For static loading at the flex arm load point in X Y Z directions:

- Stresses

- . Maximum in Upper Face
- . Maximum in Lower Face
- . Maximum in Core Webs
- . Stress in Face Sheets at .020" Deflection
- . Overall Stress Distribution

} Principal
Stresses

- Beam Loads

- . Plot B.M. in Flex Arm
- . Loads in Corner Pieces
- . Loads in Member 49

- Deflections

- . Soft Flex Arm Stiffness

F. Hayes
F. Hayes

A. Sheffler
A. Sheffler

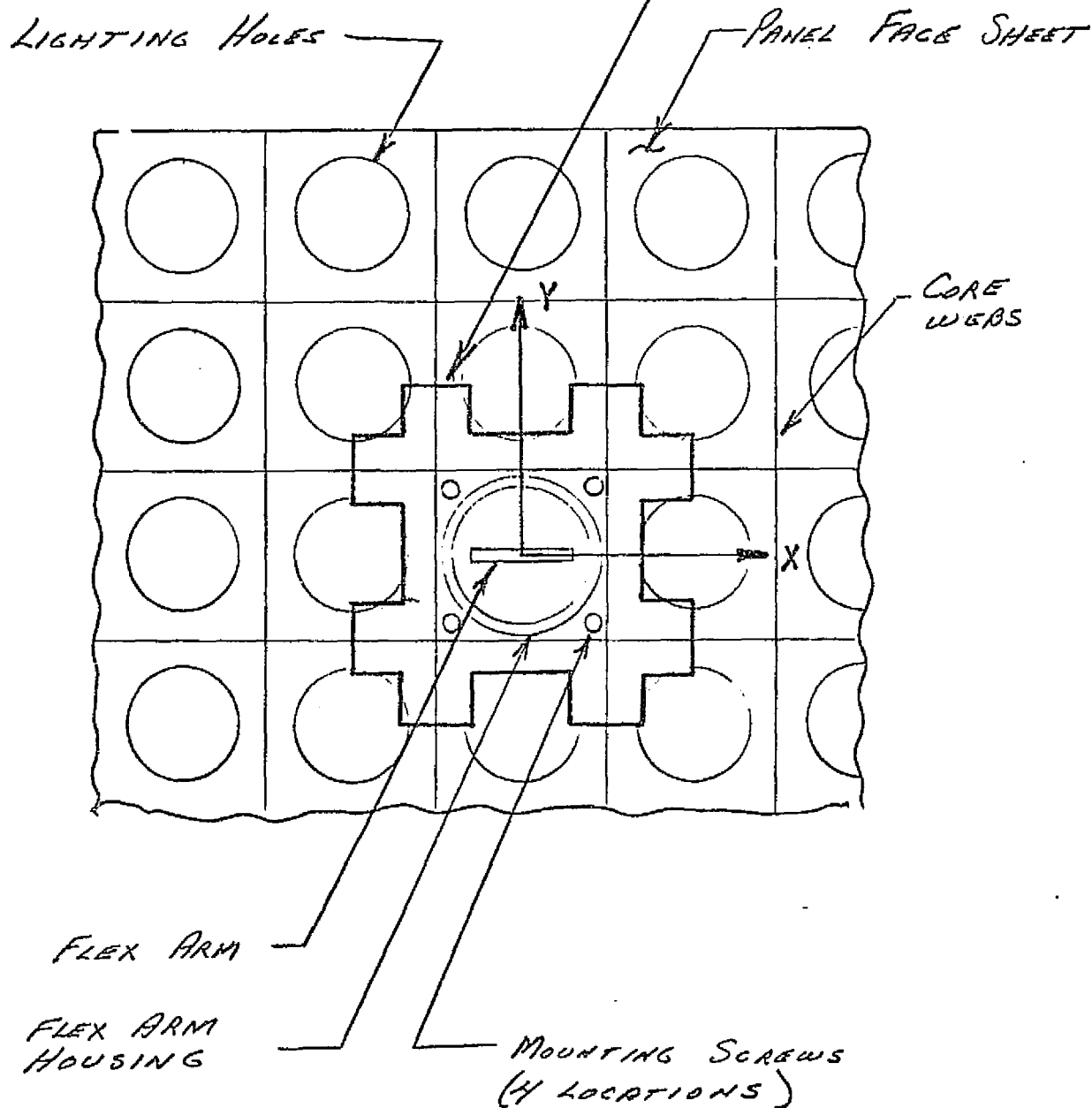
jah
Attachments

dc F. Gross
W. Martin
W. Metzger

FIGURE 1A

ILLUSTRATION OF PANEL SEGMENT
USED IN FLEX MOUNT ANALYSIS

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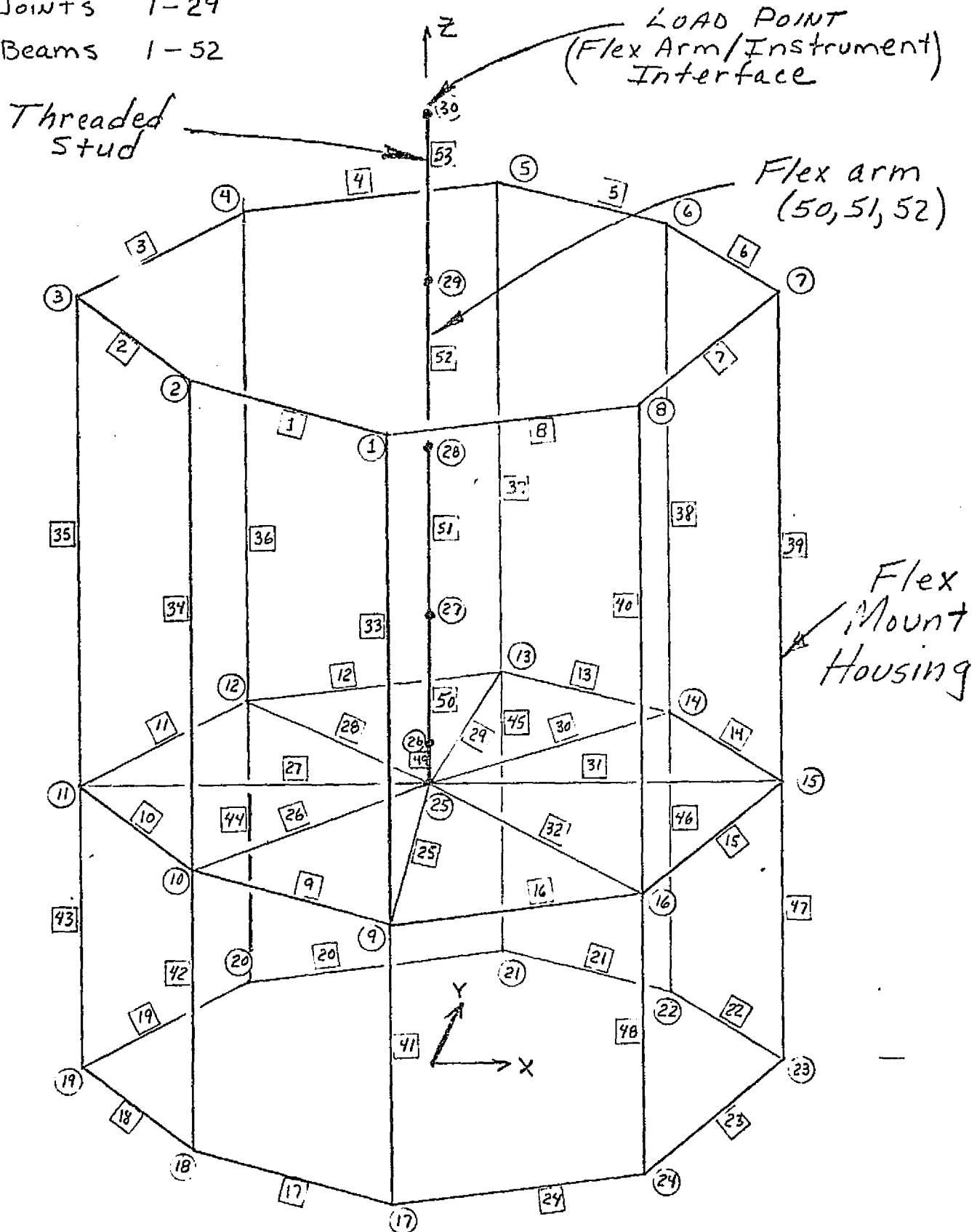
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FIGURE 1: FLEX MOUNT HOUSING

Joints 1-29

Beams 1-52



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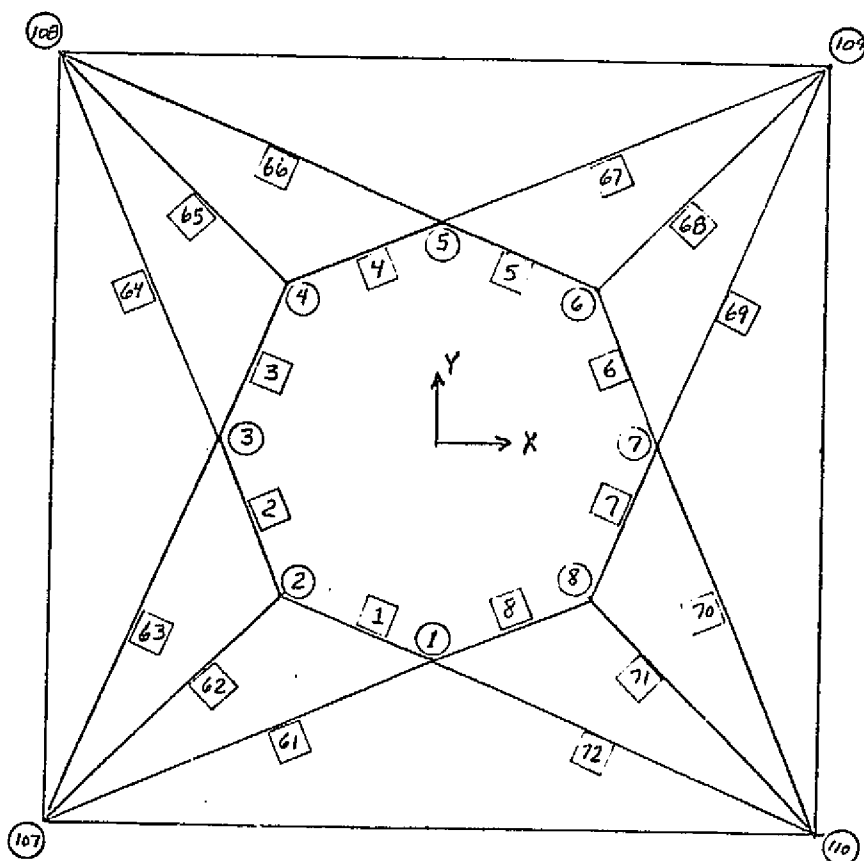
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Page 2 of 2

FIGURE 2: TOP PLATE
(Plate modeled as beams)

Beams 61-72



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FIGURE 3: Bottom Plate

Beams 81-88

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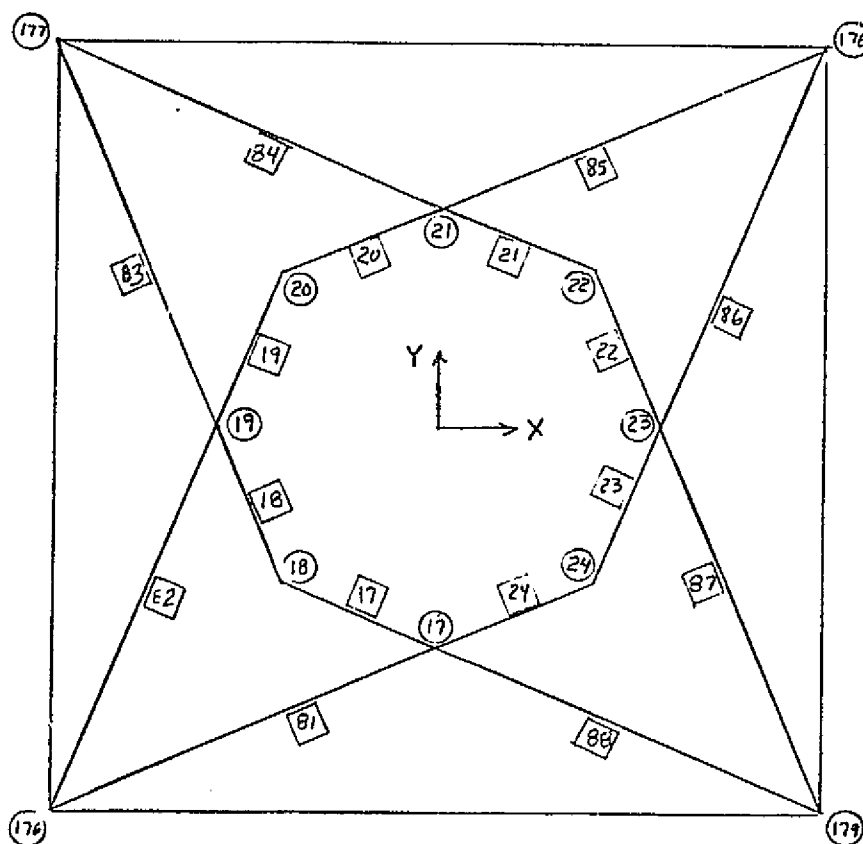


FIGURE 4

FINITE ELEMENT GRID MODEL OF I.S.P. Segment

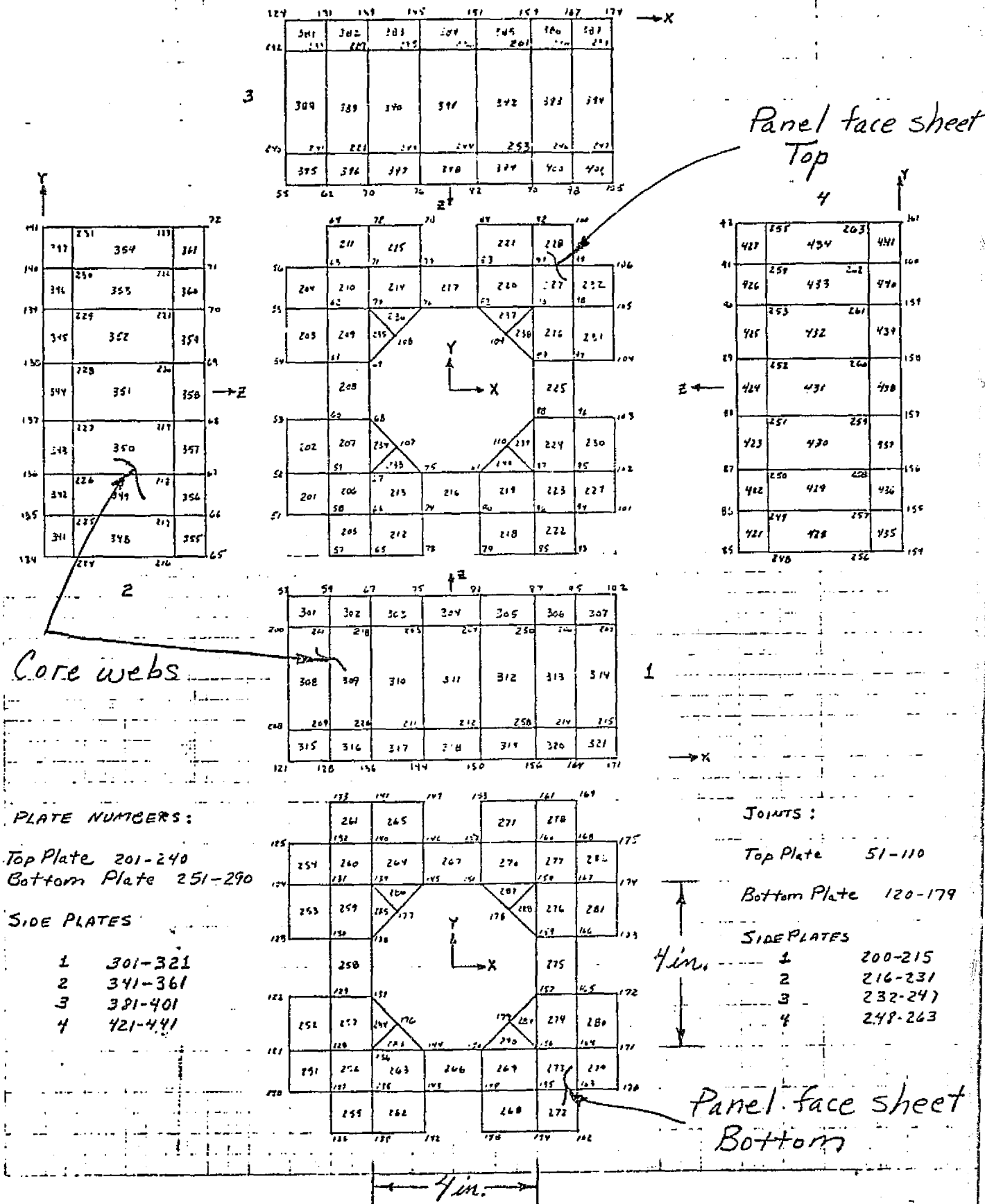


FIGURE 5

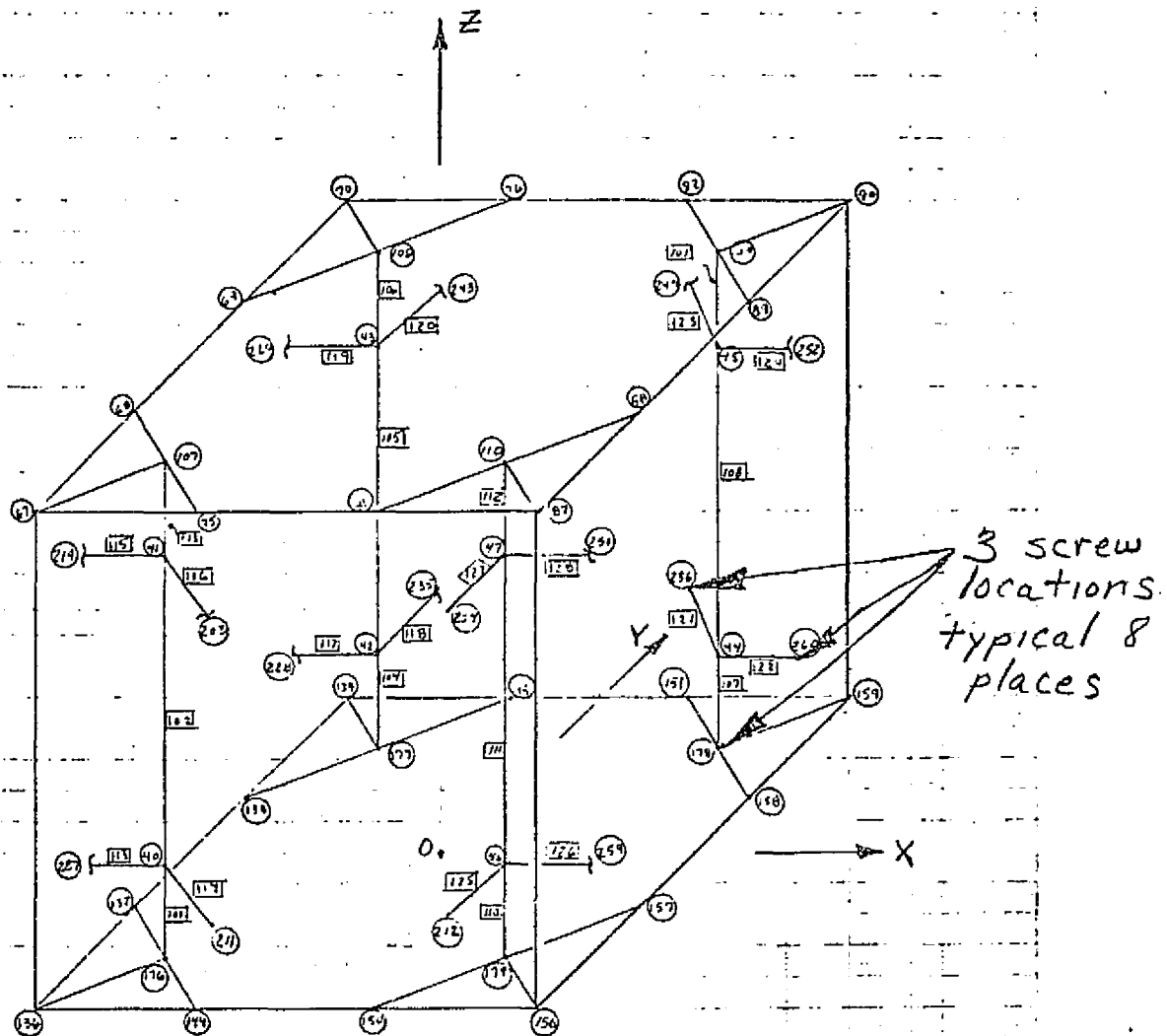
BEAMS:

101-112

DOUBLER REINFORCEMENT BEAMS (ON BOLT & 2 MA OF DOUBLER)

113-128

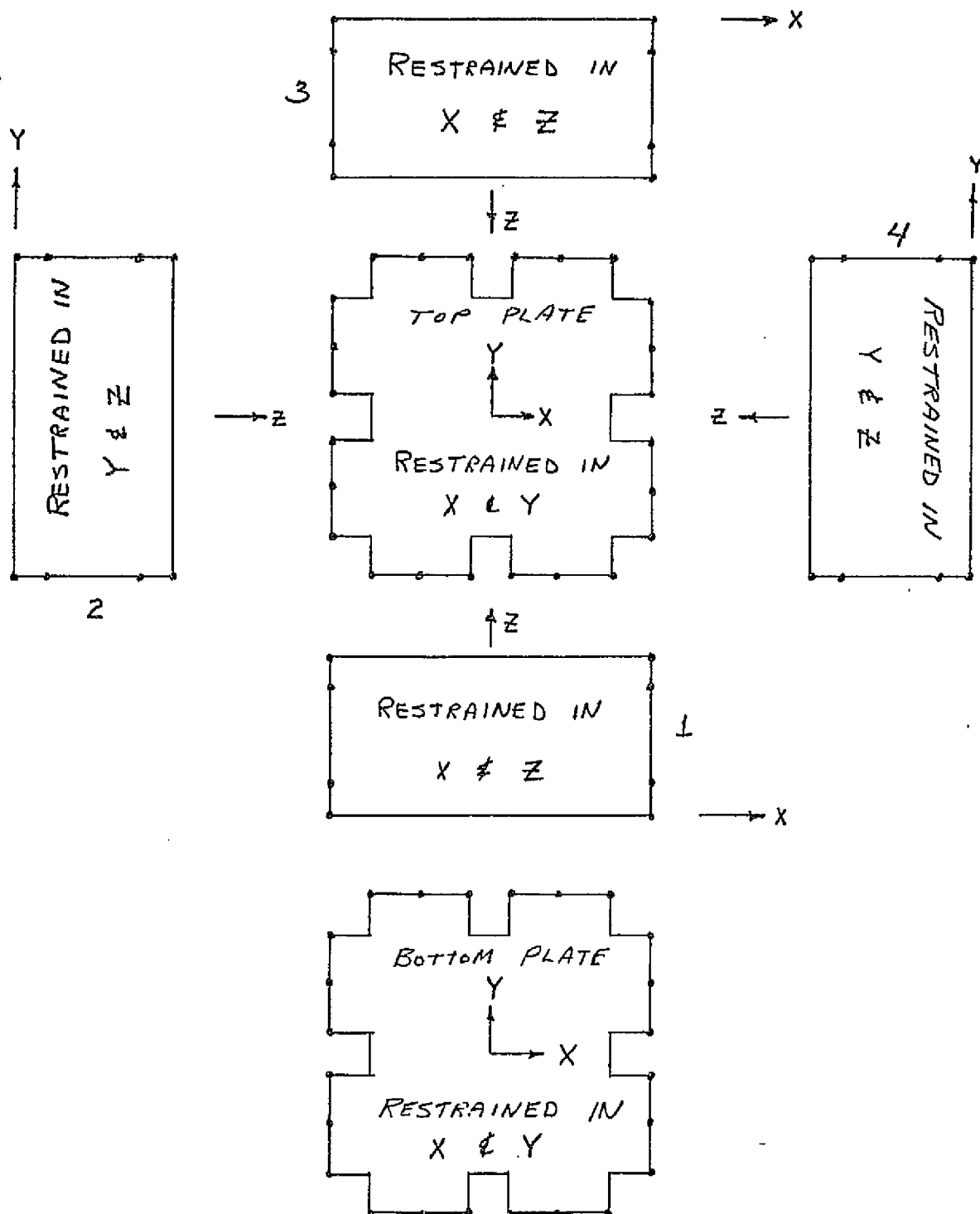
BEAMS CONNECTING DOUBLERS TO INTERCELL CORE WALLS



JOINTS: 40-47

BEAMS: 101-128

FIGURE 6: RESTRAINT SYSTEM



NOTES:

1. RESTRAINTS LOCATED AT JOINTS SHOWN

2. ALSO, JOINT 30 IS RESTRAINED FROM ROTATIONS (see Figure 1)

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FIGURE 7

MATERIAL CONSTANTS

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MATERIAL	E (lb/in ²)	G (lb/in ²)	ν
(2024-T6) Aluminum	10.5×10^6	3.95×10^6	.33
(ZK60) Magnesium	6.5×10^6	2.41×10^6	.35
(Ti 6Al) Titanium	16.0×10^6	6.1×10^6	.31
(6061-T6) Aluminum	9.9×10^6	3.72×10^6	.33

A. T. Electronics

Division

New York City

SECTION PROPERTIES 8

BEAM SECTION PROPERTIES

BEAM TYPE	MEMBERS	A(IN ²)	I _x (IN ⁴)	I _y (IN ⁴)	I _z (IN ⁴)	K _y	K _z
1	1-8	.23	.0021	.0278	.0097	.56	.54
2	9-16	.25	.05	.1	.0030	-	1.
3	25-32	.339	.0243	.00465	.0196	1.	.5
4	33-40	.11	.05	.0072	1.439×10^{-4}	-	1.
5	41-48	.11	.05	.0072	1.439×10^{-4}	-	1.
6	61-72	.094	.05	2.725×10^{-4}	.0019	1.	-
7	49	.276	.171	.0561	.1145	5.	5.
8	101-112	.167	2.185×10^{-4}	.0314	.0314	.5	.5
9	113-128	.80	.05	.05	.05	1.	1.
10	50	.55	.0146	.0036	.1742	1.	-
11	51	.423	.0112	.0028	.0793	1.	-
12	52	.31	.0082	.0021	.0313	1.	-
13	53	.11	.0019	.0010	.0010	.75	.75
14	17-24	.141	.05	.0016	.0016	.85	.85
15	81-88	.141	.05	4.087×10^{-4}	.0066	1.	-

NOTE: PROPERTIES IN LOCAL COORDINATE SYSTEM

FIGURE 9: FLEX ARM ANALYSIS

LOAD CONDITION: LATERAL Y

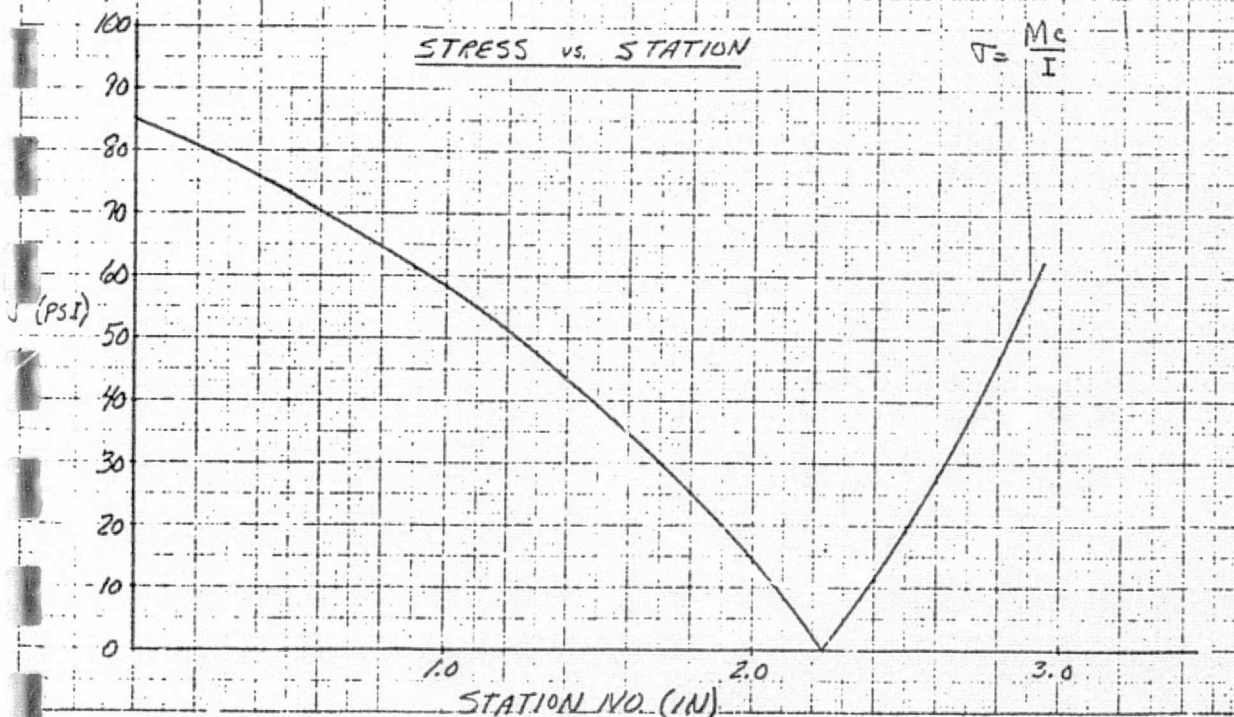
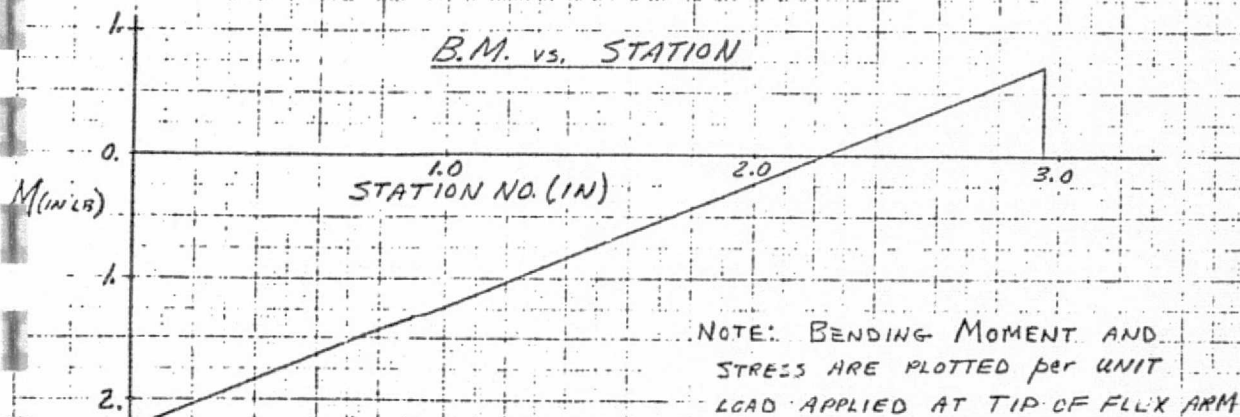
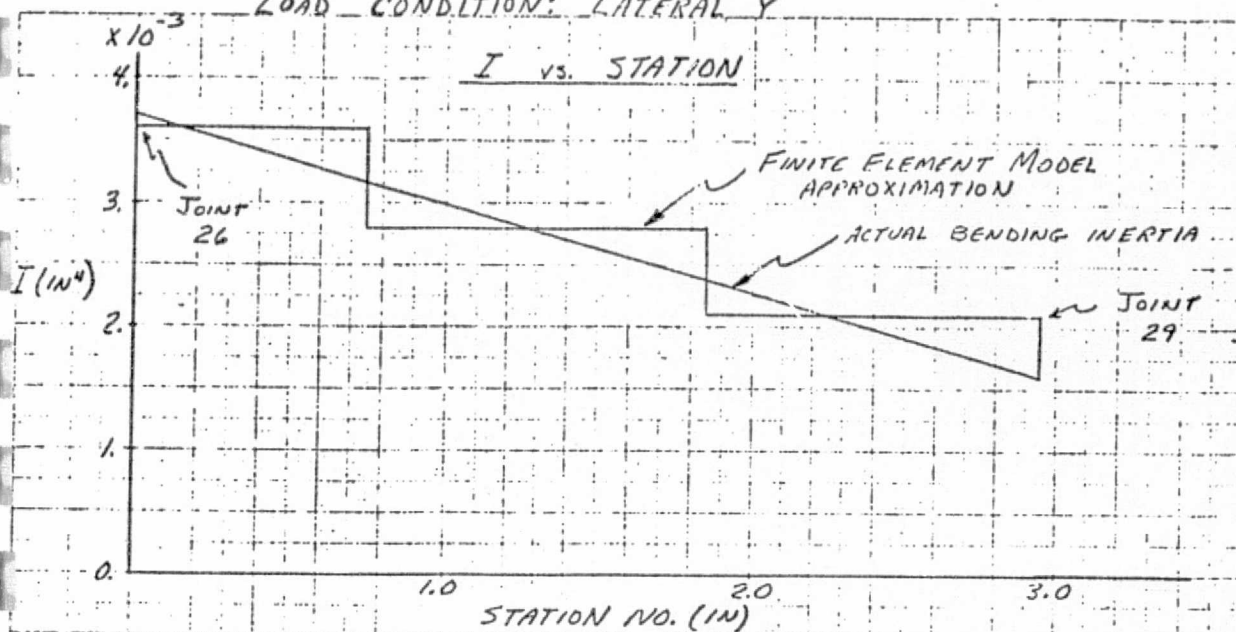


FIGURE 10: FLEX ARM ANALYSIS

LOAD CONDITION: LATERAL X

10.

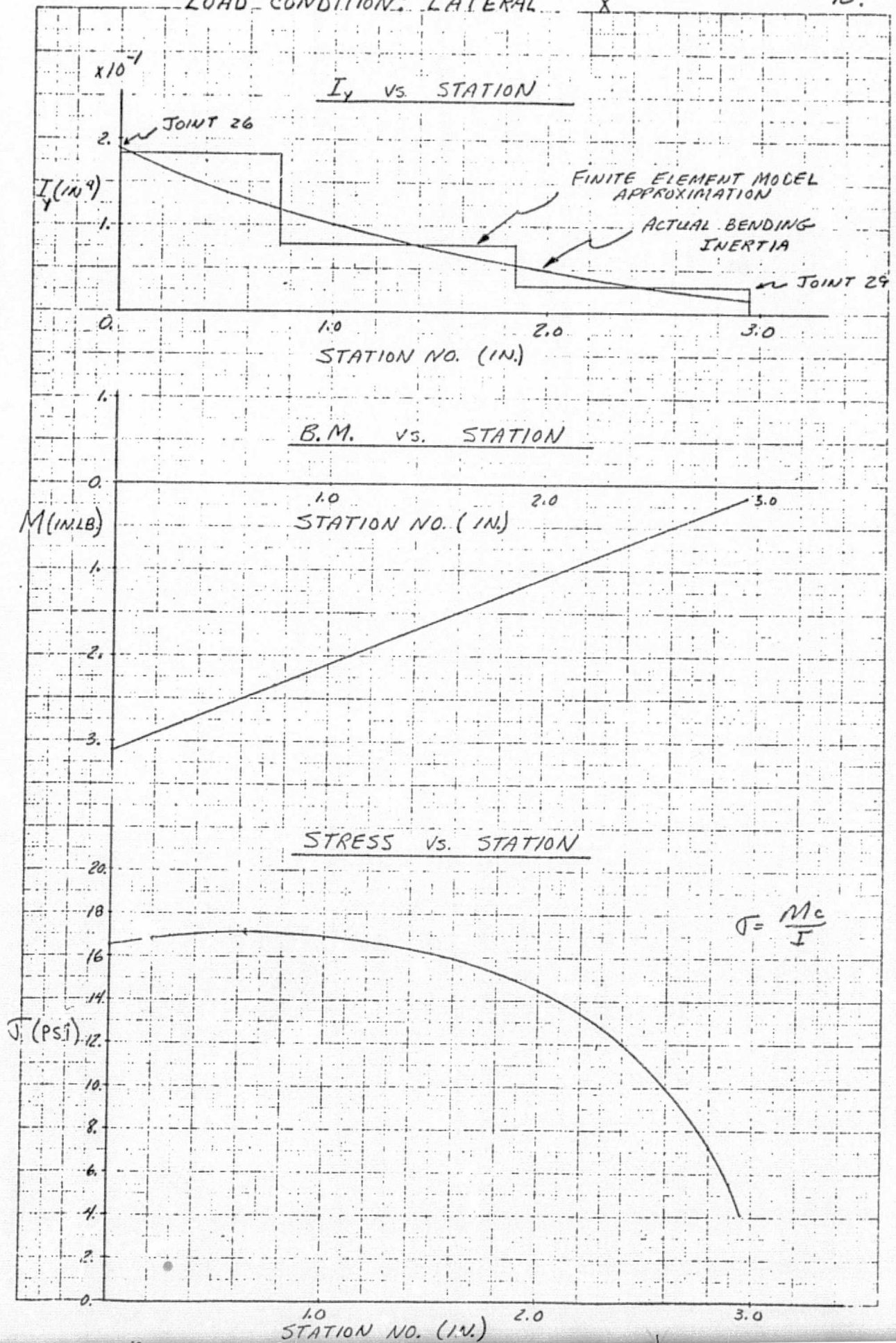
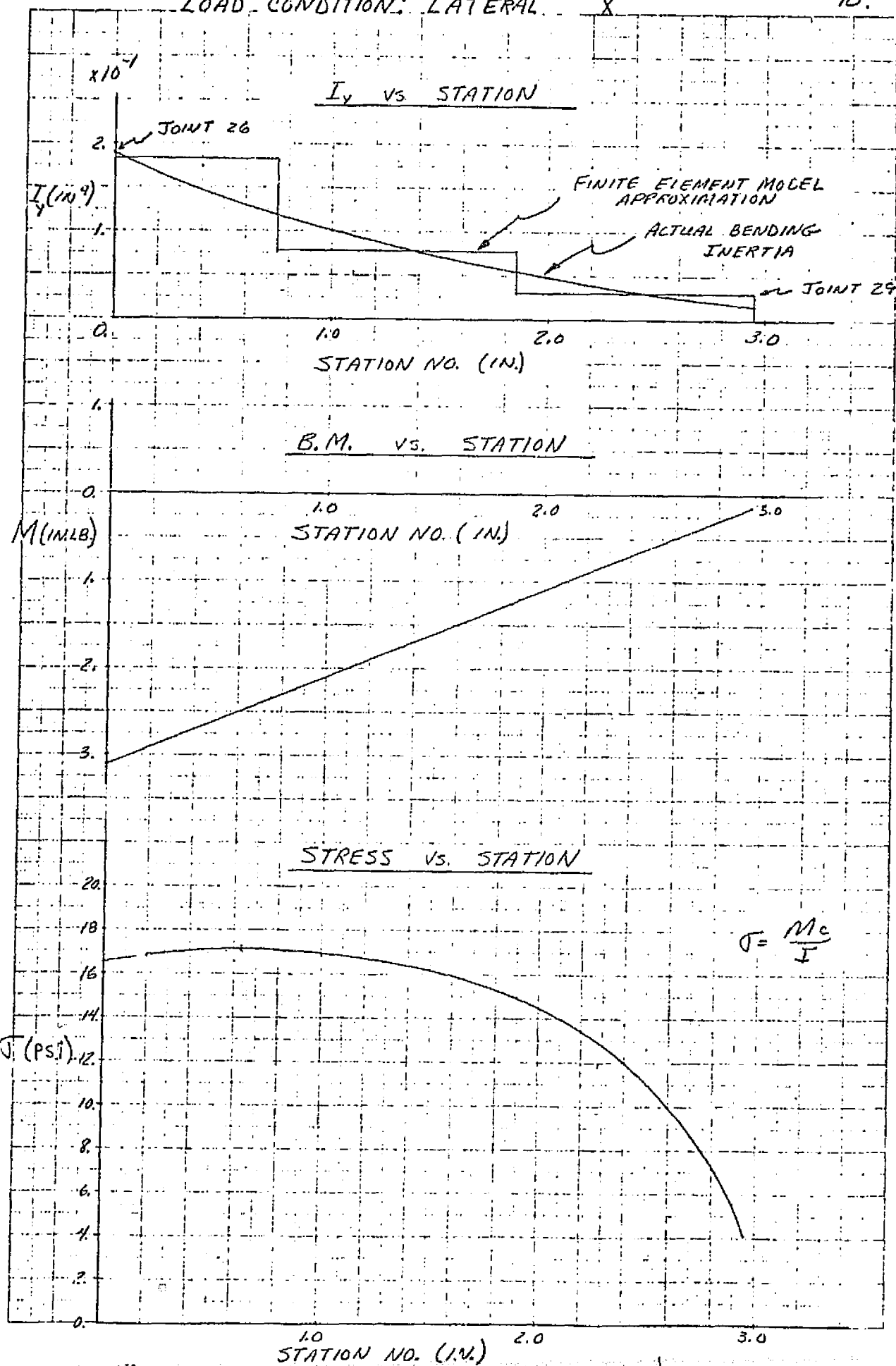


FIGURE 10: FLEX ARM ANALYSIS

LOAD CONDITION: LATERAL X

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Princeton, New Jersey

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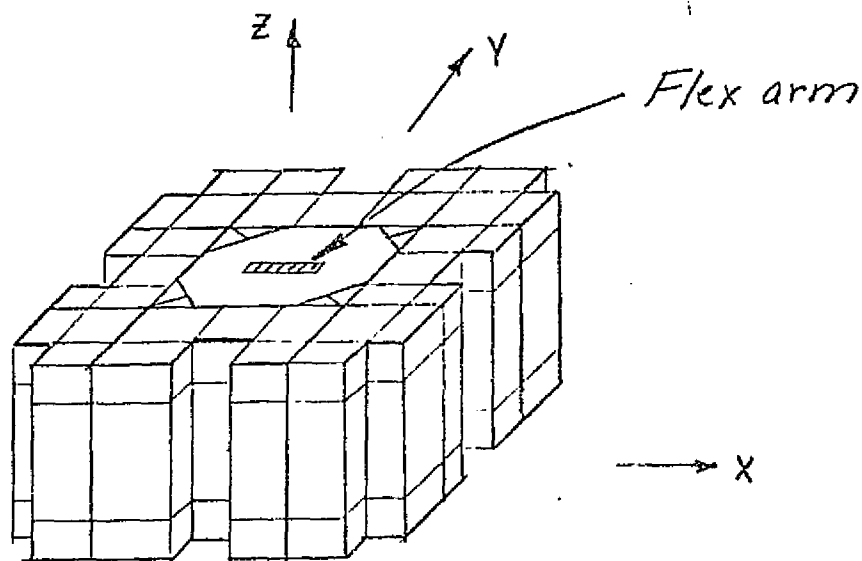
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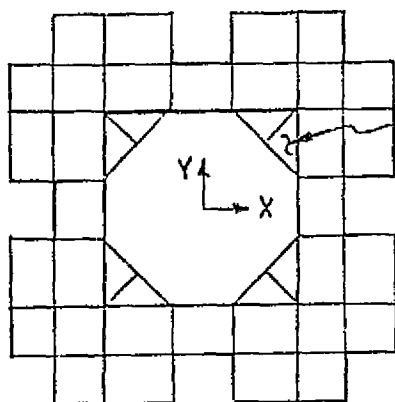
Page No. 11 of

FIGURE 11:
STRESSES IN TOP AND BOTTOM PLATES
(per unit load applied at flex mount)



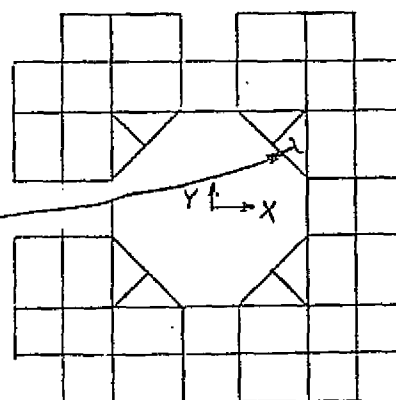
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TOP PLATE



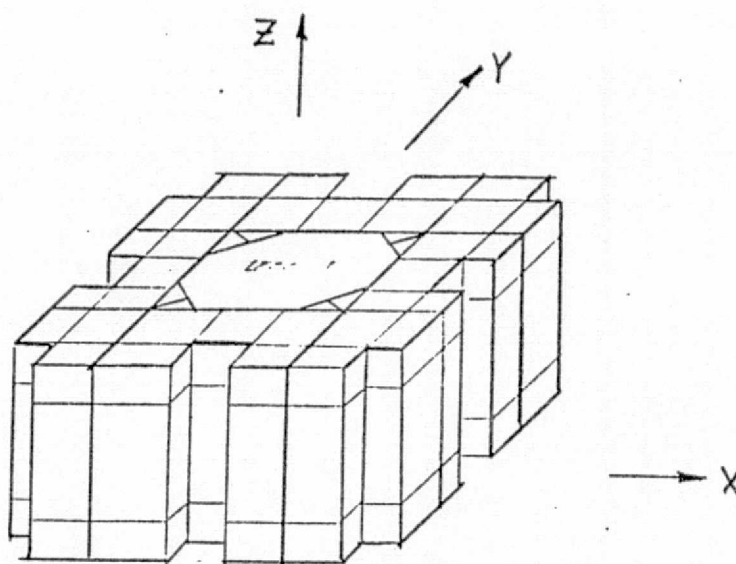
$\sigma_{MAX} = 3.224 \text{ psi}$

BOTTOM PLATE



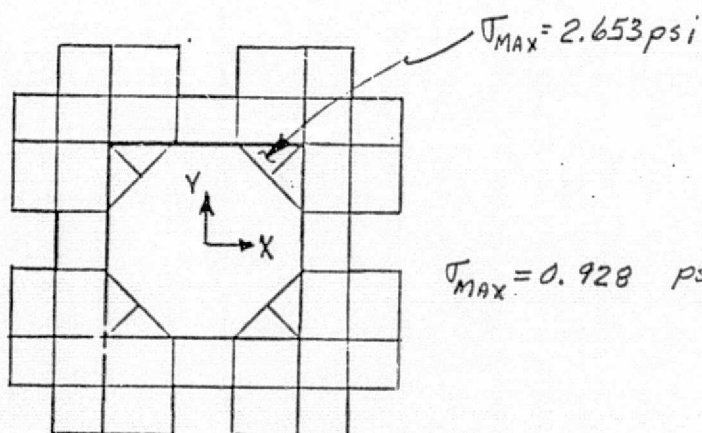
$\sigma_{MAX} = 0.527 \text{ psi}$

FIGURE 12:
STRESSES IN TOP AND BOTTOM PLATES
(per UNIT LOAD APPLIED AT FLEX MOUNT)

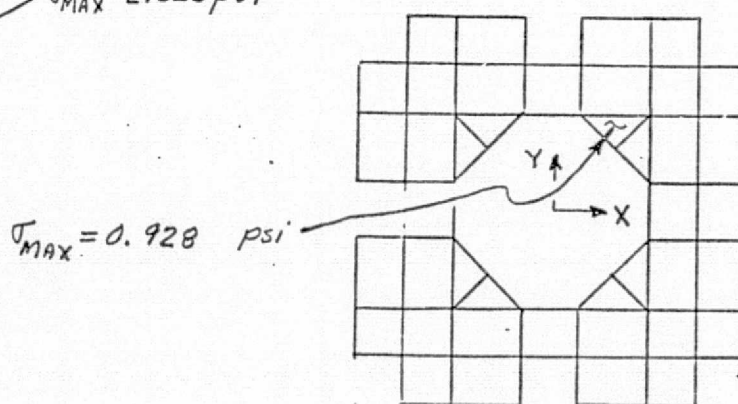


LOAD CONDITION: LATERAL Y

TOP PLATE



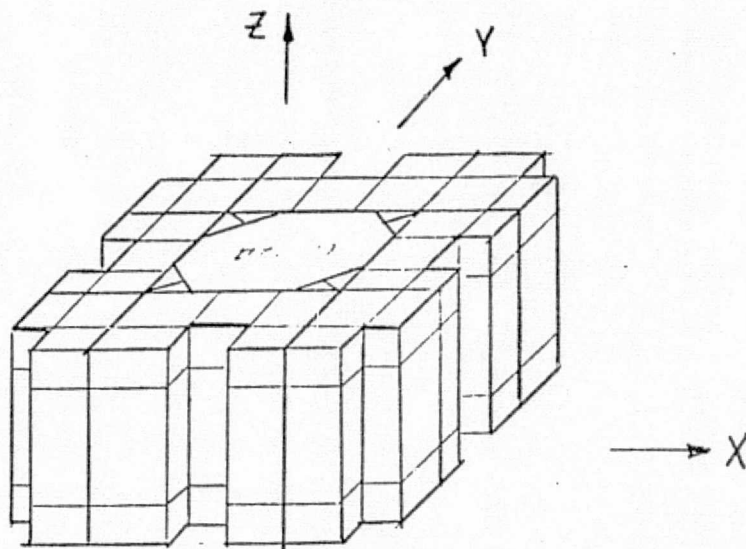
BOTTOM PLATE



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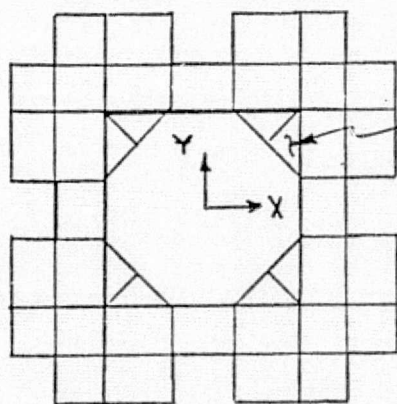
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Approved: _____ Project: SMM
Date: _____ Contract No. _____
Subject: LOAD CONDITION - AXIAL Z Page No. 13 of _____

FIGURE 13:
STRESSES IN TOP AND BOTTOM PLATES
(per UNIT LOAD APPLIED AT FLEX MOUNT)



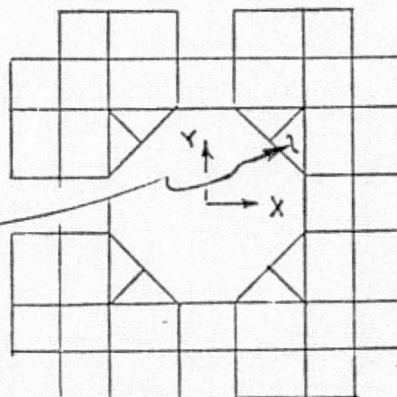
LOAD CONDITION: AXIAL Z

TOP PLATE



$$\tau_{MAX} = 0.3468 \text{ psi}$$

BOTTOM PLATE



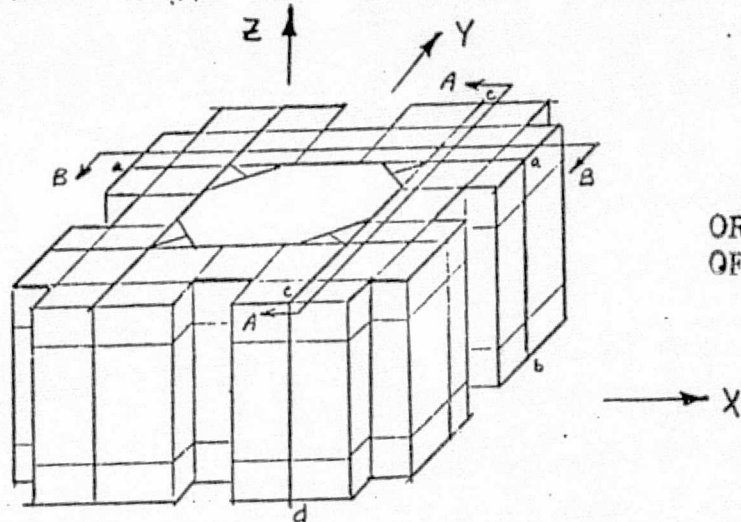
$$\tau_{MAX} = 0.2130 \text{ psi}$$

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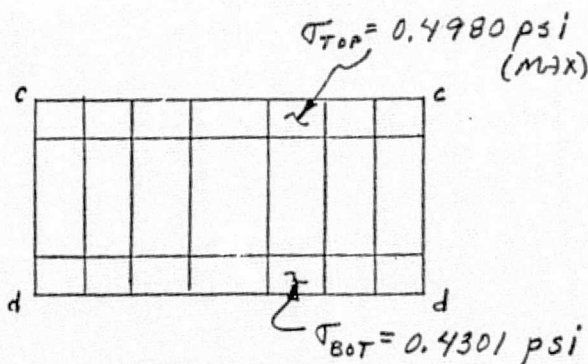
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FIGURE 14:

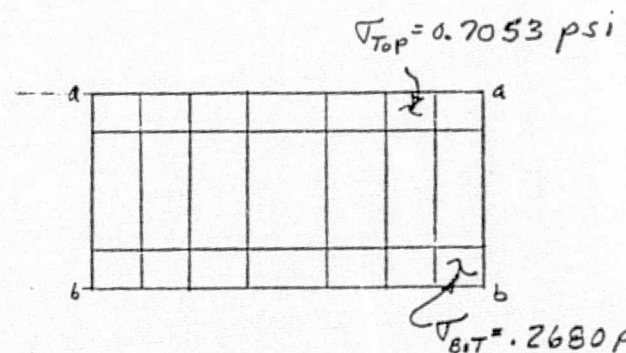
STRESSES IN INTERCELL CORE WEBS
(per UNIT LOAD APPLIED AT FLEX MOUNT)



LOAD CONDITION: LATERAL X



SECTION A-A

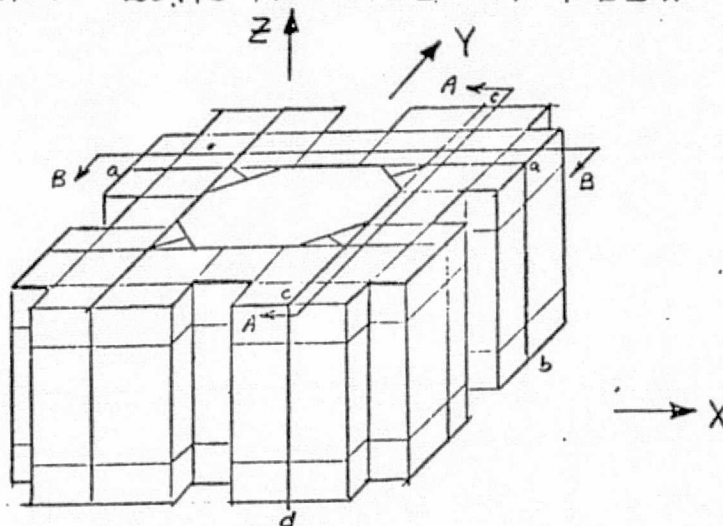


SECTION B-B

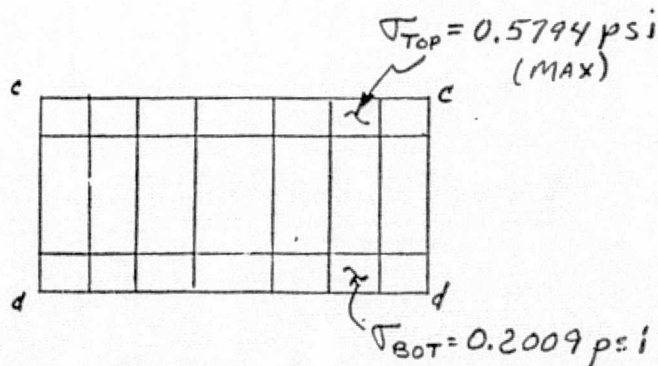
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 Fort Worth, Texas

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 Date: _____ Contract No. _____
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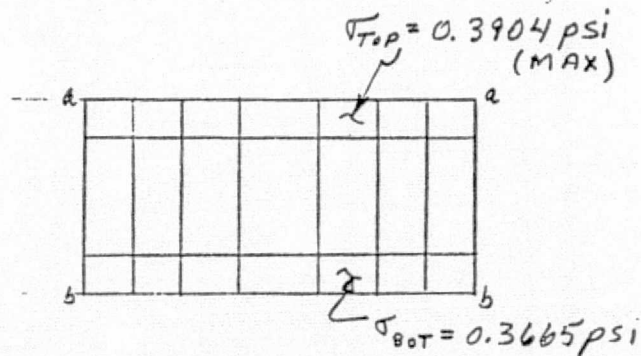
FIGURE 15:
STRESSES IN INTERCELL CORE WEBS
 (per UNIT LOAD APPLIED AT FLEX MOUNT)



LOAD CONDITION: LATERAL Y



SECTION A-A

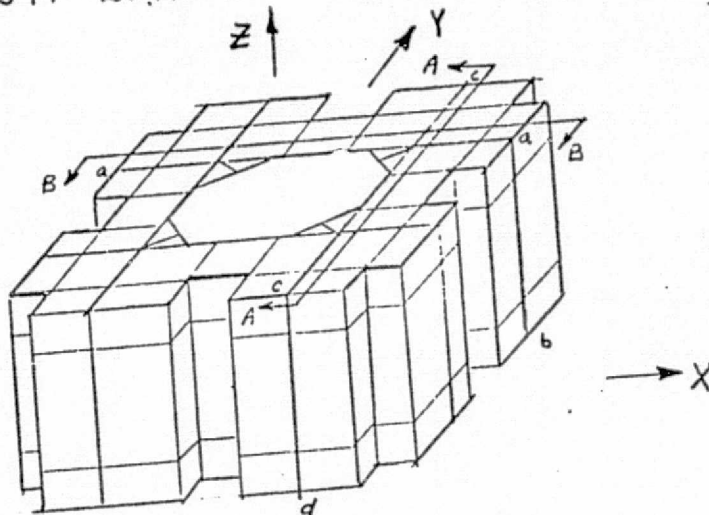


SECTION B-B

LOAD CONDITION: AXIAL Z

FIGURE 16:

STRESSES IN INTERCELL CORE WEBS
(per unit load applied at FLEX MOUNT)



LOAD CONDITION: AXIAL

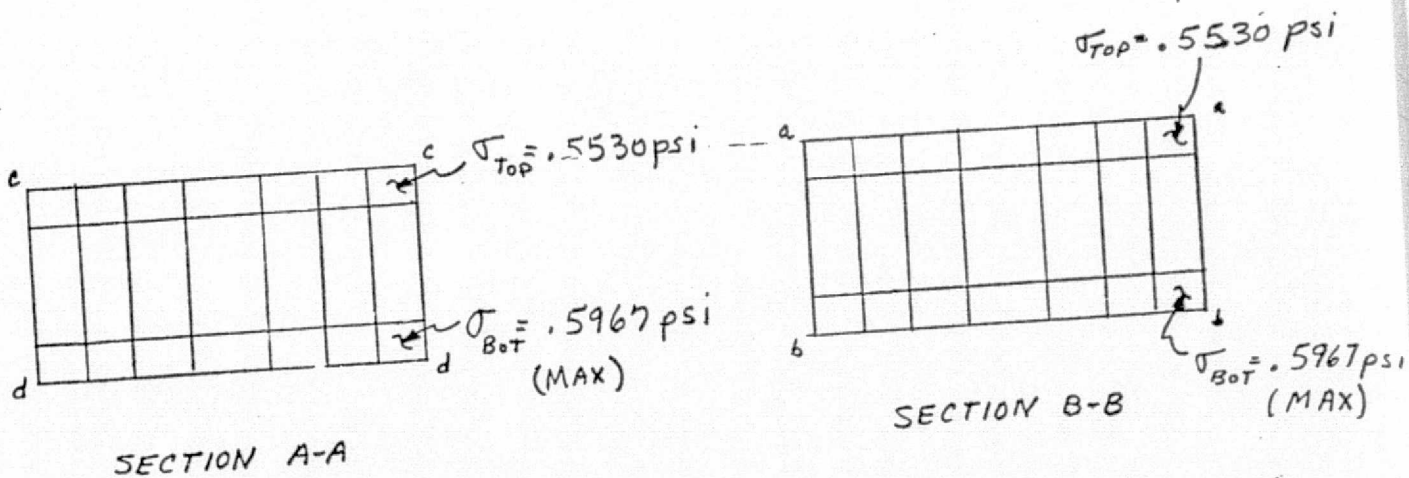


FIGURE 17:

STRESS IN FACE SHEETS @ .020" DEFLECTION

SINCE 0.020" DEFLECTION CORRESPONDS TO A
LATERAL SHEAR LOAD OF 141.0 LB, THE
MAXIMUM STRESS IN THE FACE SHEETS IS:

TOP PLATE:

$$\tau_{MAX} = (141.0)(2.653) = \underline{\underline{374.07 \text{ psi}}} *$$

BOTTOM PLATE:

$$\tau_{MAX} = (141.0)(0.928) = \underline{\underline{130.85 \text{ psi}}}$$

INTERCELL CORE WALLS:

TOP:

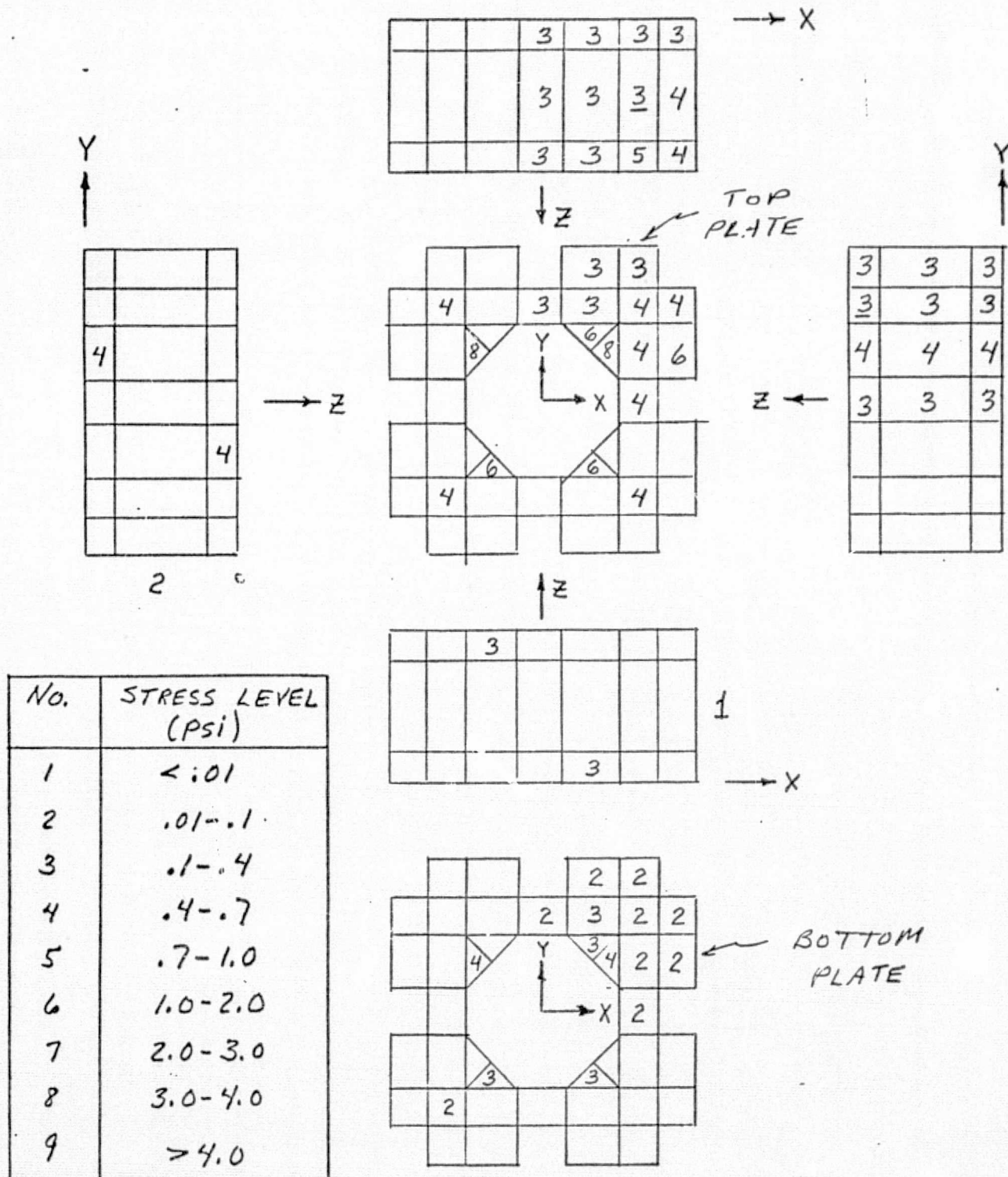
$$\tau_{MAX} = (141.0)(.5794) = \underline{\underline{81.70 \text{ psi}}}$$

BOTTOM:

$$\tau = (141.0)(0.3665) = \underline{\underline{51.68 \text{ psi}}}$$

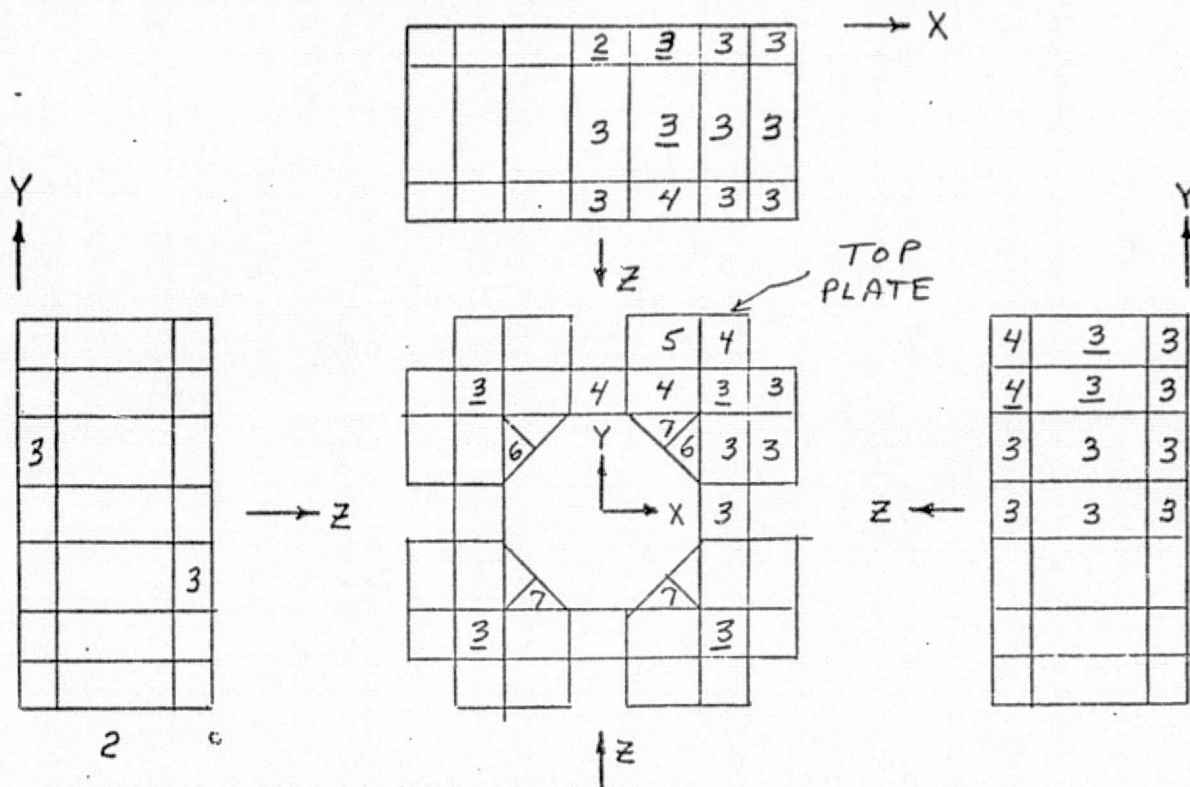
* Exceeds 100 psi design requirement

FIGURE 18: STRESS DISTRIBUTION IN PLATES

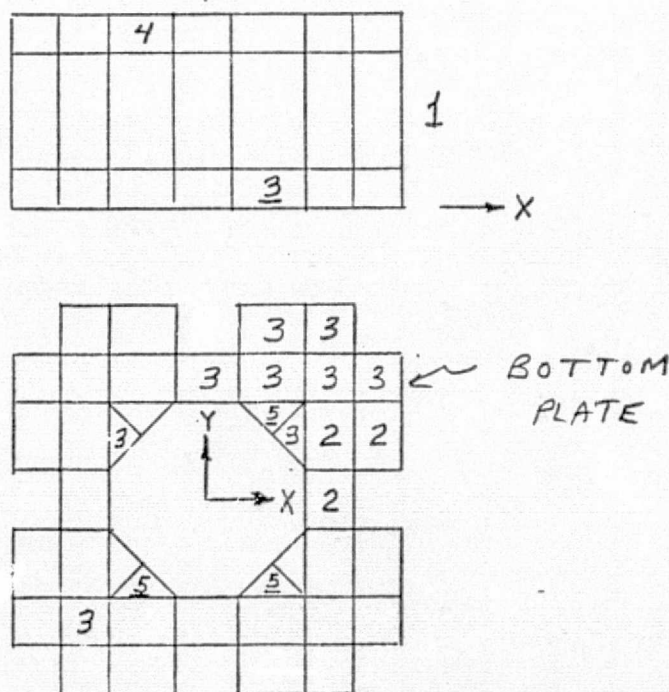


LOAD CONDITION: LATERAL X

NOTE: STRESSES per UNIT LOAD APPLIED AT FLEX MOUNT
ALL STRESSES ARE PRINCIPAL STRESSES



No.	STRESS LEVEL (psi)
1	< .01
2	.01 - .1
3	.1 - .4
4	.4 - .7
5	.7 - 1.0
6	1.0 - 2.0
7	2.0 - 3.0
8	3.0 - 4.0
9	> 4.0



NOTE: STRESSES per UNIT LOAD APPLIED AT FLEX MOUNT
ALL STRESSES ARE PRINCIPAL STRESSES

ASTRO

Astro Electronics
Division

Princeton, New Jersey

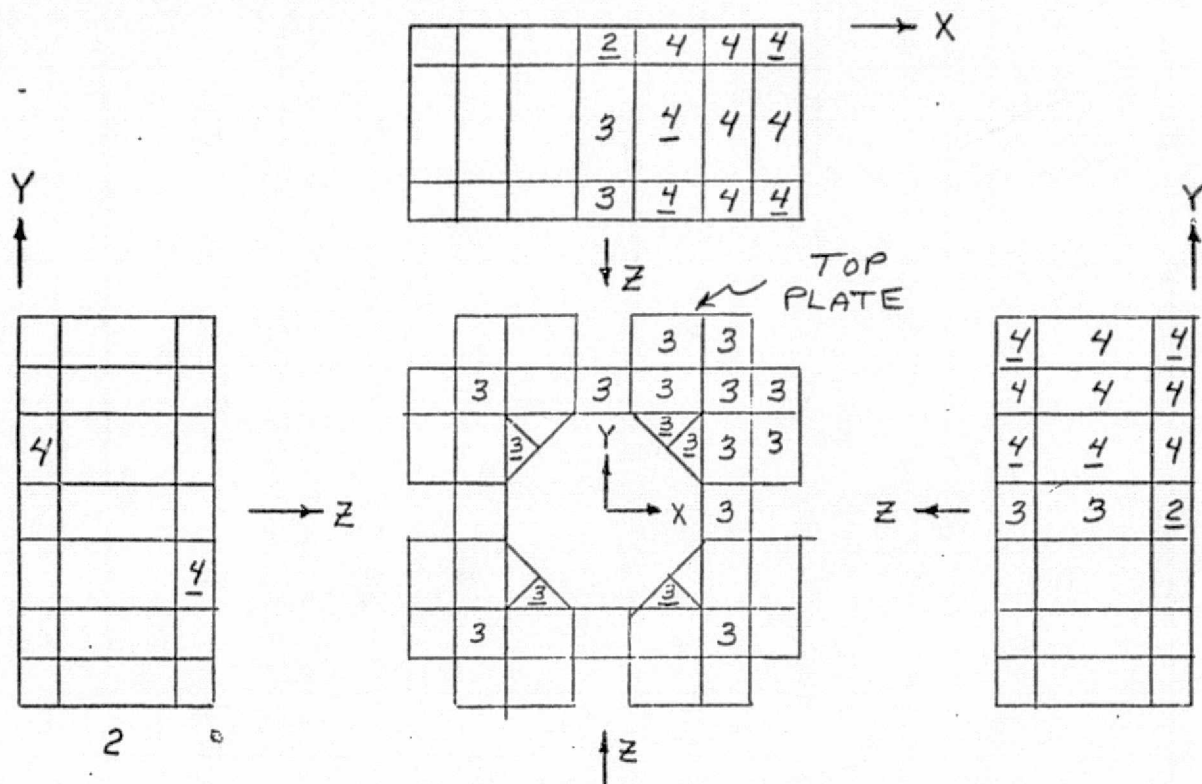
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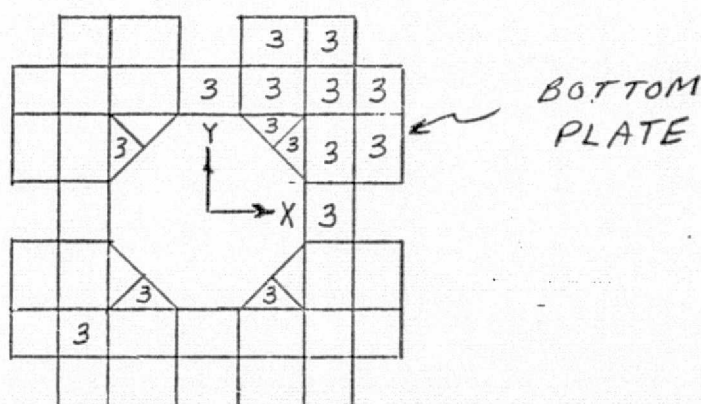
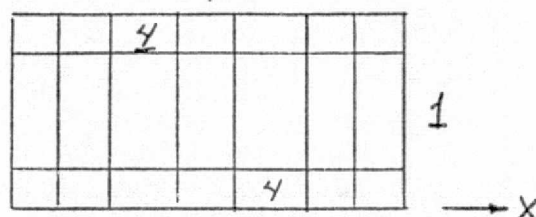
Date: _____ Contract No. _____

Subject: LOAD CONDITION: AXIAL Z Page No. 20 of _____

FIGURE 20: STRESS DISTRIBUTION IN PLATES



No.	STRESS LEVEL (Psi)
1	<.01
2	.01-.1
3	.1-.4
4	.4-.7
5	.7-1.0
6	1.0-2.0
7	2.0-3.0
8	3.0-4.0
9	>4.0



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OF POOR QUALITY

LOAD CONDITION: AXIAL Z

NOTE: STRESSES per UNIT LOAD APPLIED AT FLEX MOUNT
ALL STRESSES ARE PRINCIPAL STRESSES

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Date: 6/19/75

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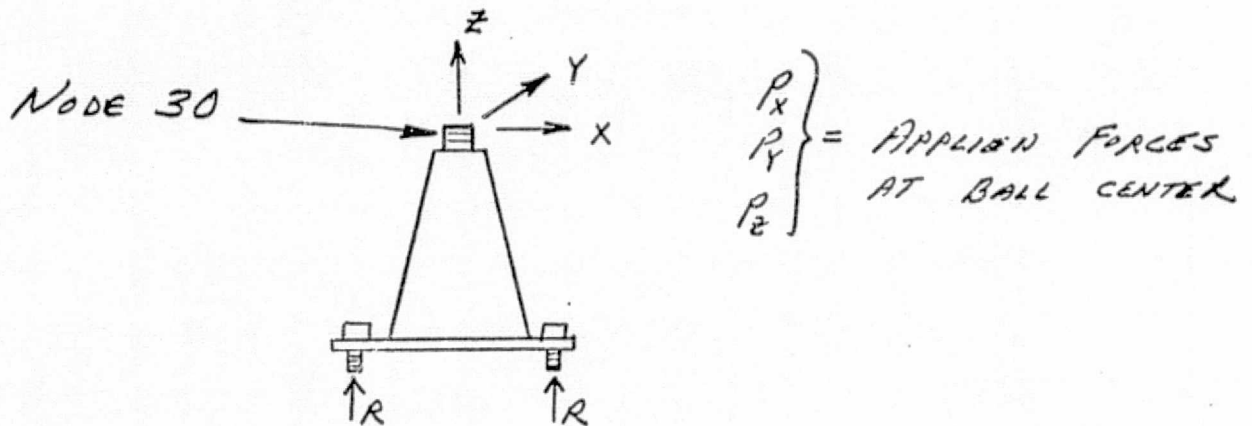
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FIGURE 21:
AXIAL LOADS IN FLEX PARM MOUNTING SCREWS

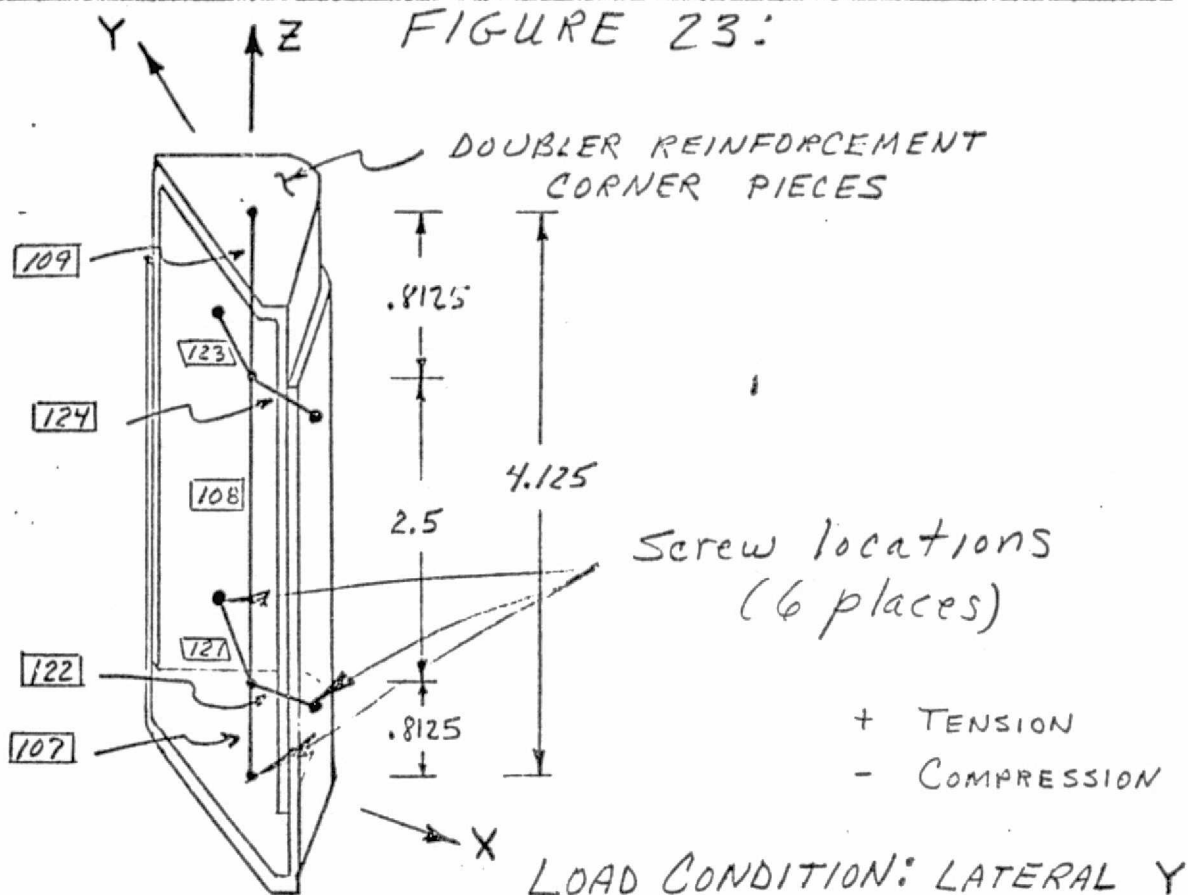


$R = \text{AXIAL LOAD IN MOUNTING SCREWS}$

FROM FEM ANALYSIS:

$$R = .40(3.098)P_x + 1.14(2.238)P_y + .25P_z$$

$$\therefore R = \underline{\underline{1.24P_x + 2.55P_y + .25P_z}}$$

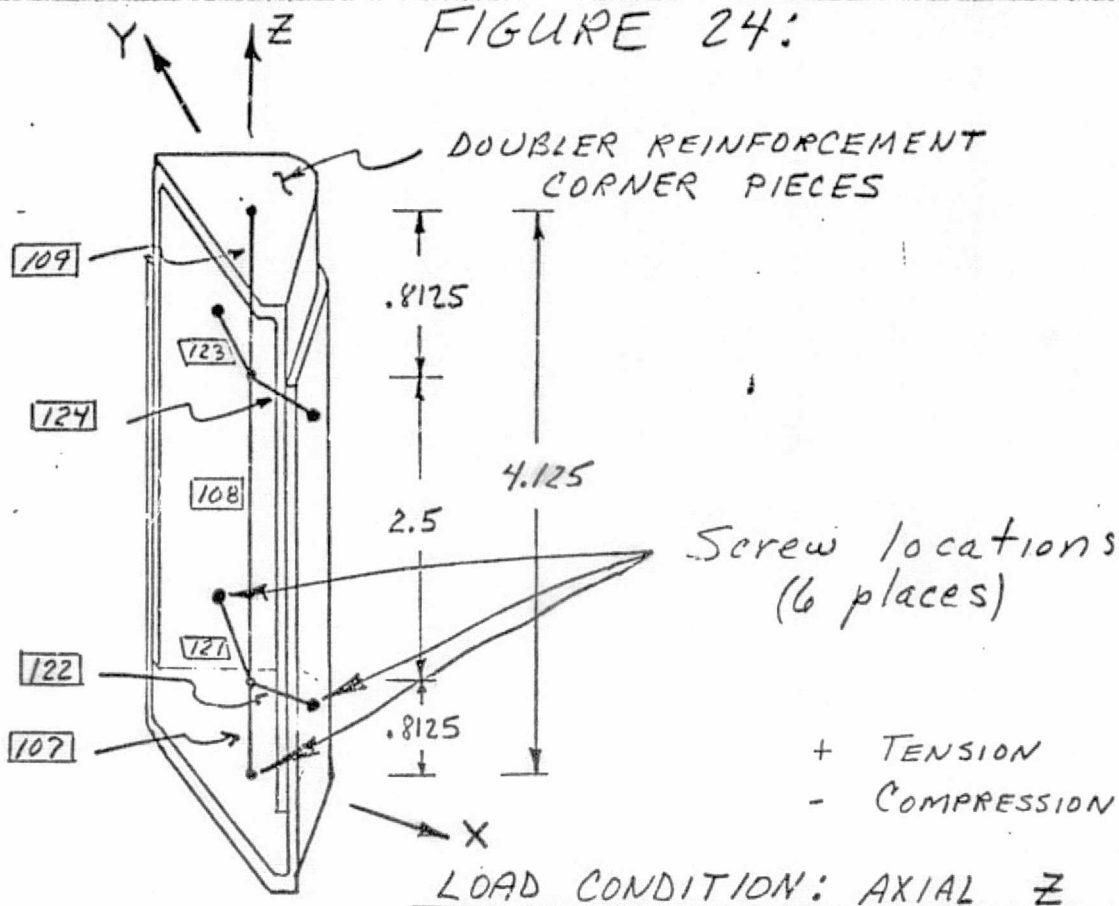


FORCES per UNIT LOAD Y

BEAM NO.	Y DIRECTION SHEAR (LB)	Z DIRECTION SHEAR (LB)	AXIAL (LB)
107	-	-	+ 0.04282
108	-	-	- 0.00163
109	-	-	- 0.05528
121	0.04436	0.00864	-
122	0.00009	0.00929	-
123	0.03007	0.01791	-
124	0.02358	0.01818	-

NOTES:

1. INTERFACE JOINTS LIE ON PLATE CENTERLINES
2. FORCES GIVEN IN LOCAL COORDINATE SYSTEM
3. UNIT LOAD APPLIED AT FLEX MOUNT



FORCES per UNIT LOAD Z

BEAM NO.	Y DIRECTION SHEAR (LB)	Z DIRECTION SHEAR (LB)	AXIAL (LB)
107	-	-	- 0.07802
108	-	-	+ 0.02235
109	-	-	+ 0.15787
121	0.05018	0.00684	-
122	0.05018	0.00684	-
123	0.06776	0.01183	-
124	0.06776	0.01183	-

NOTES:

1. INTERFACE JOINTS LIE ON PLATE CENTERLINES
2. FORCES GIVEN IN LOCAL COORDINATE SYSTEM
3. UNIT LOAD APPLIED AT FLEX MOUNT

APPENDIX D

FLEX ARMS LOADS ANALYSIS

SOLAR MAXIMUM MISSION FLEX ARMS LOADS ANALYSIS

G. VARADARAJAN

27 January 1976

TABLE OF CONTENTS

- Section 1. Description of Analysis
- Section 2. Loads Summary
- Section 3. Description of Model
- Section 4. Equivalent Panel Properties
- Section 5. Beam Section Properties
- Section 6. Computer Plots of the ISP and the Boxes

SECTION 1

DESCRIPTION OF THE ANALYSIS

1.1 Introduction. An instrument support panel, its support structure, and the simulated instrument boxes were analyzed to obtain various static and dynamic loads in the flex and fixed arms for given static and dynamic inputs. A finite element model of the entire structure was made, and the STARDYNE program was used to obtain the required loads.

1.2 -Static analysis. The four load conditions defined for the Delta Launch Vehicle are:

- a. 17.8g x +3.3g Y
- b. 17.8g x +3.3g Z
- c. 17.8g x -3.3g Y
- d. 17.8g x -3.3g Z

The flex arm loads for the above load cases are presented in Tables 1 through 4. Instrument Boxes #1 and #29 show high nodal displacements for the above load cases.

1.3 Modal analysis. Based on excessive deflections found in static analysis, two modifications were made in the geometry of the structure before carrying out the modal analysis. Additional beam elements were added in these two boxes to stiffen them and reduce the oscillations in the modal analysis.

The first mode is a bending mode of Box #16 at 7.67 Hz. The second mode at 9.82 Hz shows high amplitude nodal oscillations in the panel, but once again Box #16 shows the highest oscillations. The third mode at 10.38 Hz is clearly a panel bending mode. The fifth mode at 16.07 Hz is a panel torsion mode. The fourth mode at 14.29 Hz shows high displacements at nodes 547, 548, and 549 of Box #1 (see Figure 7) even though the rest of the box does not. This may be avoided by stiffening the baseplate of Box #1. The higher modes are mostly of the boxes, many of them showing high displacements in Box #16. The two panel mode shapes are shown in Figures 15 and 16. Figures 17 and 18 show the mode shapes of the pedestal and the side strut.

Mode	Frequency (Hz)	Description
1	7.67	Box #16 Bending.
2	9.82	Box #16 Bending. High Panel Oscillation Too.
3	10.38	Panel Bending Mode.
4	14.29	High Displacements at Nodes 547, 548, and 549 of Box #1.
5	16.07	Panel Torsion Mode.
Higher Modes		Mostly Box Modes. Many Showing High Displacements in Box #16.

1.4 Harmonic analysis. The following are the dynamic inputs to the fixed base ISP model:

- a. 1g X Base input.
- b. 1g Y Base input.
- c. 1g Z Base input.

The loads in the flex and fixed arms for these inputs were obtained for these 1g base inputs. These loads are scaled such that for a given input at the base of the spacecraft, the loads in the flex arms can be obtained.

Also, the given input to the base of the spacecraft is notched at peak response frequencies so as not to exceed the allowable load level in the adapter members at the base of the spacecraft. (NASA has provided the static and dynamic analyses of the ISP on the modular spacecraft.) All the loads in the flex arms for a base input of 1g X, Y, or Z to the structure are scaled to a different value of the base input such that the acceleration at the top corner of the ISP does not exceed the value obtained due to the notched input at the base of the spacecraft. These loads are given in Tables 5, 6, and 7.

The notched input values at the base of the spacecraft for the peak response frequencies and the peak responses of the top corner of the panel for these inputs are given in Table 8.

The calculations of beam section properties are given in Section 3. Computer plots of the panel and the boxes are given in Section 4.

1.5 Weight and center of gravity data. Total weight of the structure and the instrument boxes: 1773.80 lbs.

Center of gravity of the structure:

X = 66.53 inches

Y = 0.43 inches

Z = 3.09 inches

1.6 Summary and recommendations. The large deflections found in the panel, the supporting structure, and the instrument boxes were attributed to the following:

- a. "Soft" ball support beam (No. 579) of the fixed support--By doubling the moment of inertia of this beam, the deflections were reduced only by about 6 percent in the Z direction, for a 10g load. Structural changes in the entire fixed support may be necessary to solve this problem.
- b. "Soft" platform structure under the pedestal side support--This is the main cause for the low frequency modes in the structure. Presently, one end of the base of the pedestal side support is resting at the center of the platform structure. It is suggested that the pedestal be fixed at all four corners for the test configuration.
- c. "Soft" plate mounting of instrument box #1--By stiffening this plate, the deflections at nodes 547, 548, and 549 (Figure 7) can be reduced to a large extent.
- d. "Soft" structure of instrument box #16 (mode shapes shown in Figures 19 and 20)--However, structural changes may not be required in that box for strengthening it. A better distribution of mass over the modes of that box in the model thereby bringing the C.G. of the box closer to the panel would increase the frequency of that box mode.

Also, by including the effect of the 3" x 3" bar running along the base plate and the three other bars inside base box No. 1 of instrument #16 in the model would stiffen up the present model thereby increasing the frequency of that box mode.

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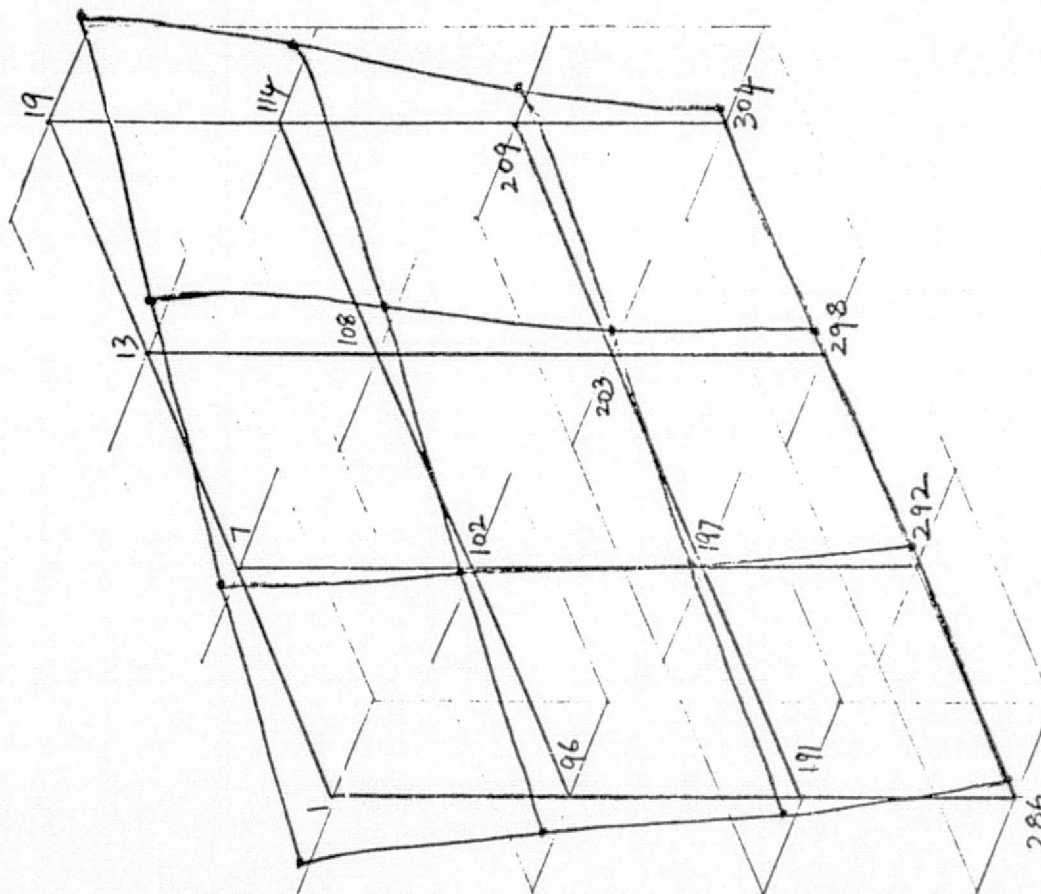
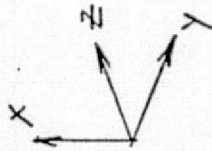
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FREQUENCY = 16.07 Hz

X DEFLECTIONS NOT SHOWN (-01--03)

FIGURE 15

PROB/7

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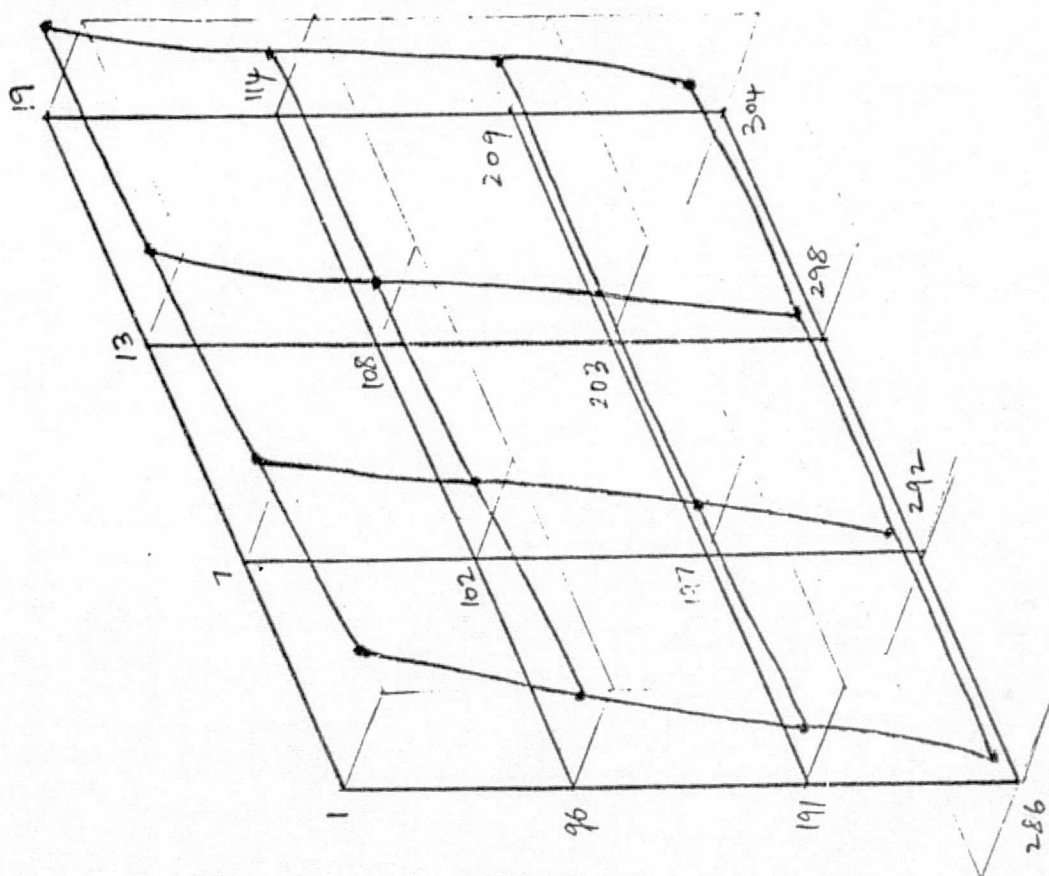
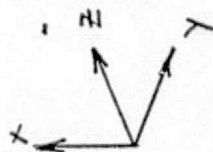
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FREQUENCY = 10.38 HZ

X DEFLECTIONS NOT SHOWN (.03-.07)

FIGURE 16

ASTRO
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FREQ = 10.3 Hz
MODAL ANALYSIS

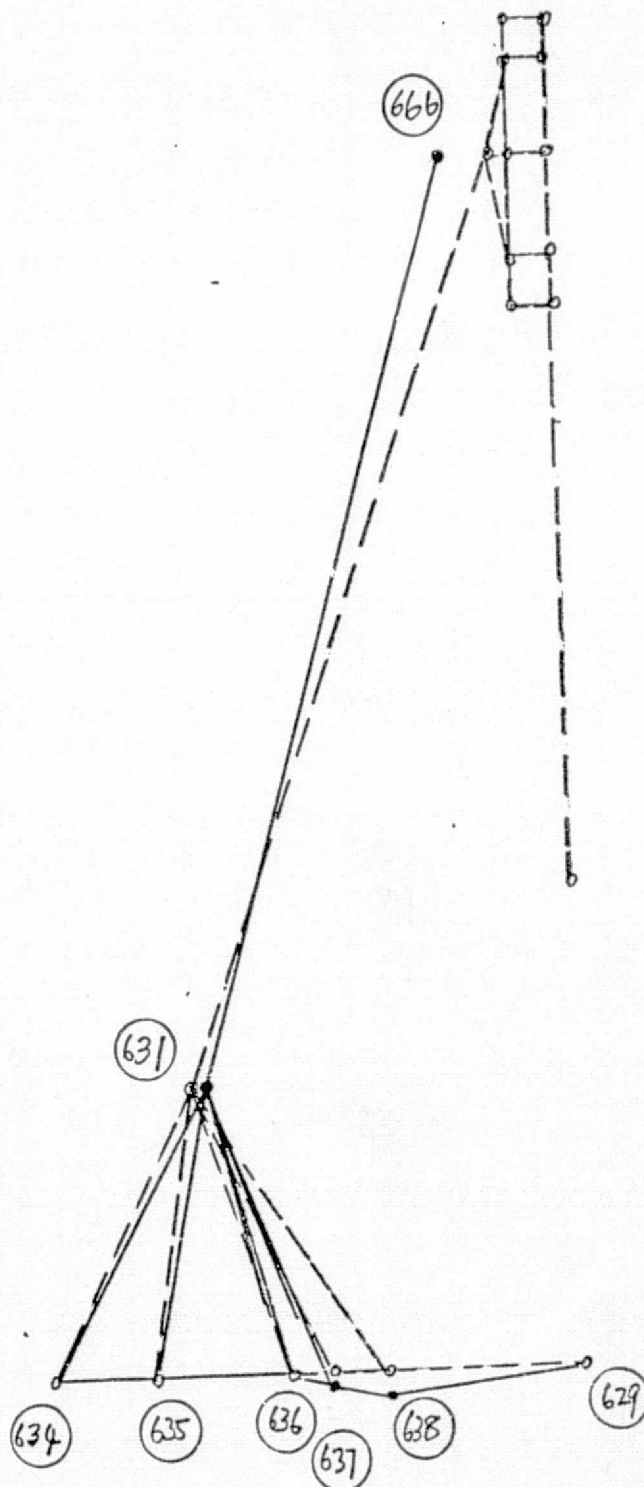


FIGURE 17

INCB/71

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FREQ = 10.3 Hz
MODAL ANALYSIS

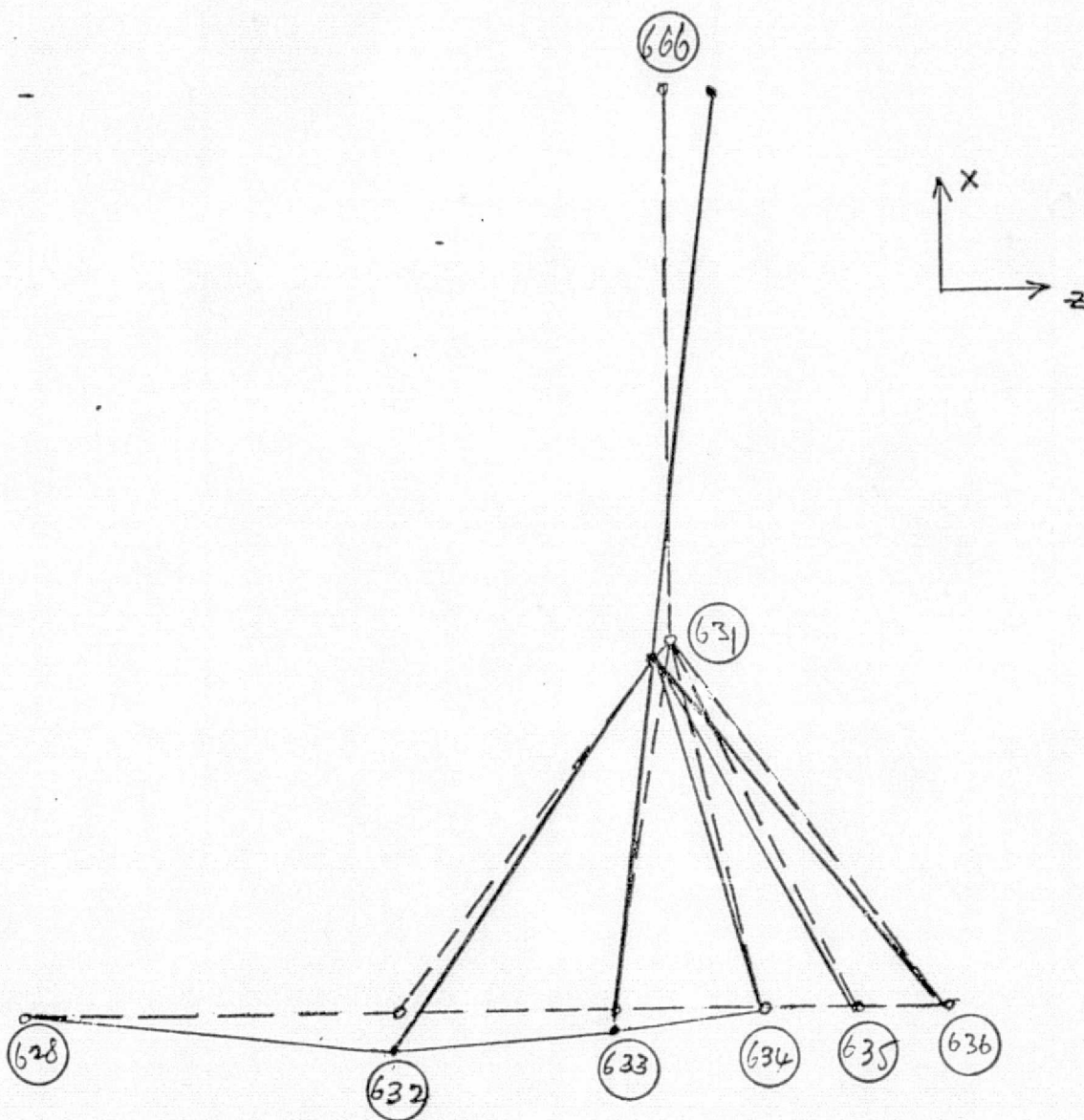
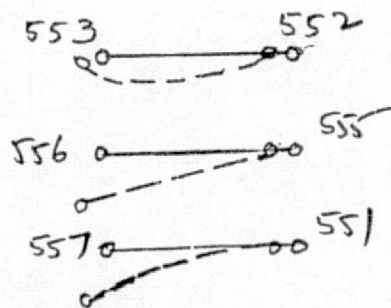
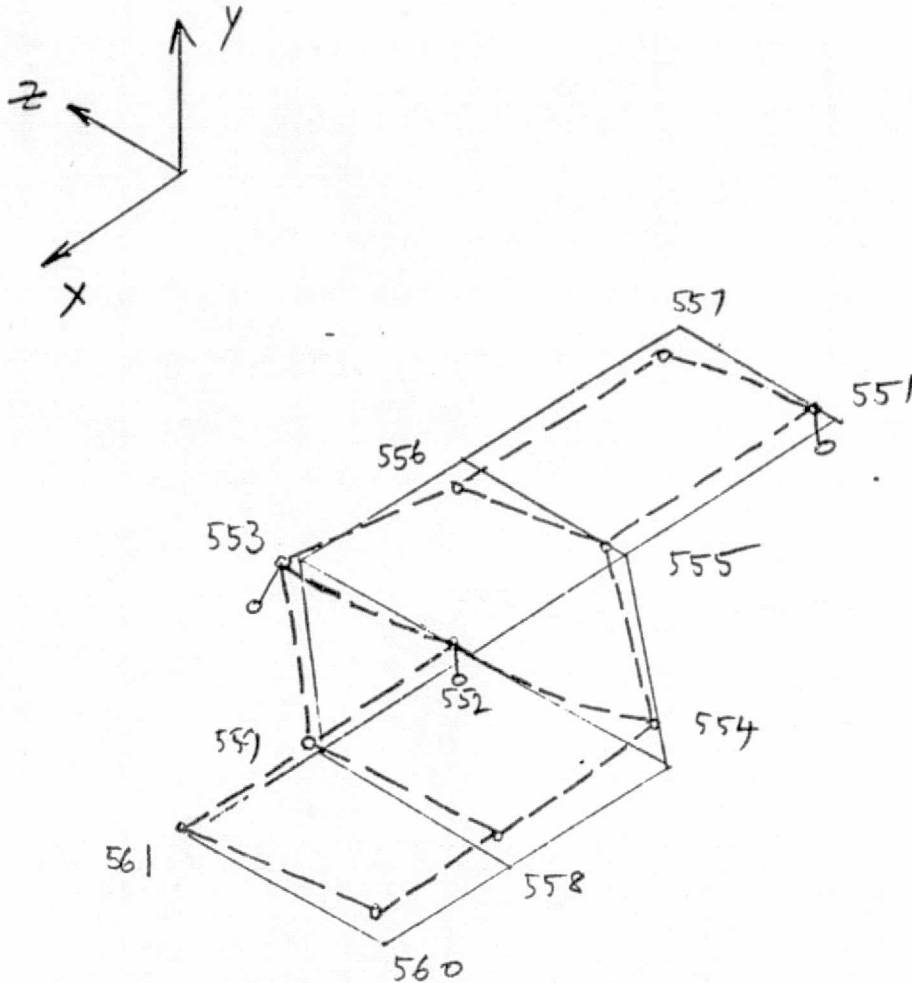


FIGURE 18

INCORPORATED

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MODE # 1
FREQ: 7.67 Hz
BASE PLATE INST. #16

FIGURE 19

URCBA

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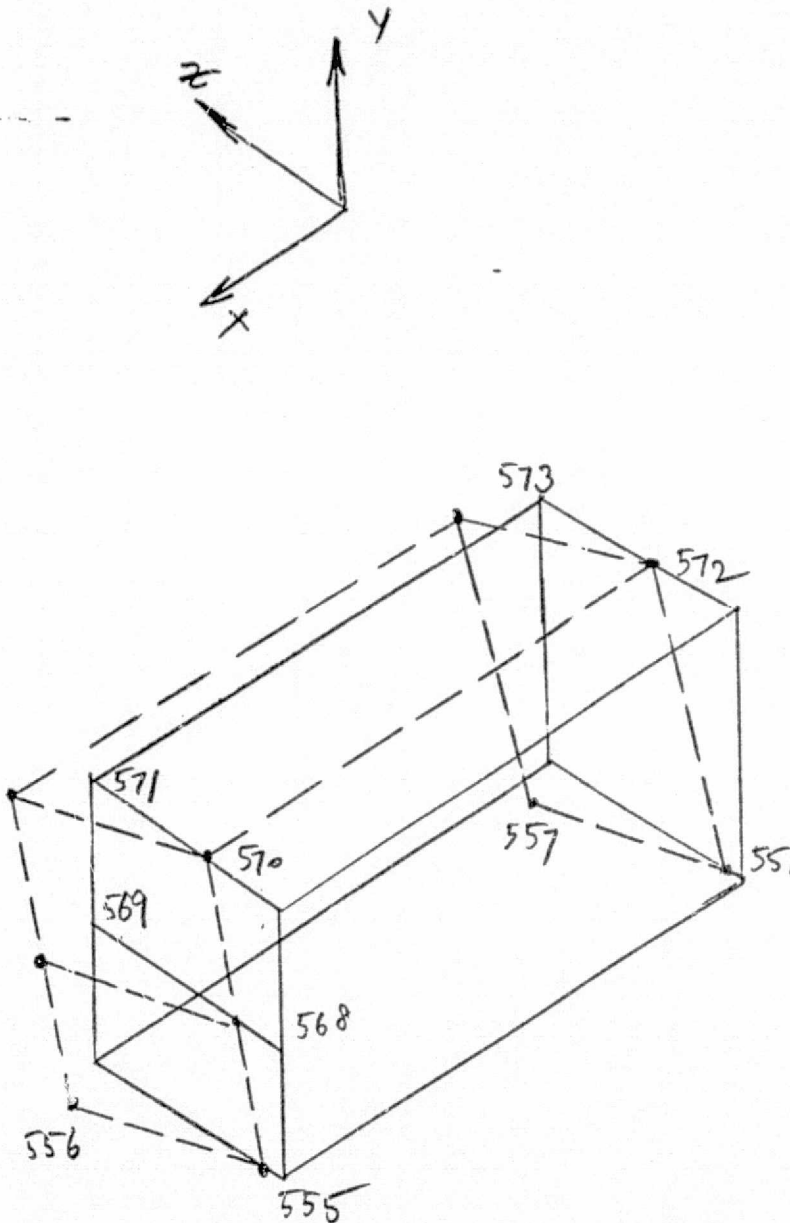
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Subject: Modal Analysis Inst. #16 Page No. ____ of ____



MODE # 1

FREQ: 7.67 Hz

PART OF INST. # 16

(BASE BOX #1)

FIGURE 20

SECTION 2

LOADS SUMMARY

TABLE 1. STATIC LOAD SUMMARY

Load Condition: 17.8g x +3.3g Y

Box No.	Attach Arm	P _X	V _Y	V _Z	M _X	M _Y	M _Z
		LBS	LBS	LBS	LB-IN.	LB-IN.	LB-IN.
25	Fixed-E2	2282	564.3	2059	117.0	3183.	-92.1.
	Flex-E1	-323.3	-85.9	-1028.	5.81	113.47	14.67
	Flex-E3	-2612	-339.7	-640.6	3.42	-1045.	543.3
16	Fixed-F2	588.9	-282.6	-2457	30.95	-3702	505.2
	Flex-F1	1.1	32.93	791.8	4.90	359.3	-54.25
	Flex-F3	-1316	-378.1	-311.3	22.4	-101.8	577.1
1	Fixed-G3	-359.7	-58.2	-1768.8	-7.67	-1661	298.7
	Flex-G1	-234.5	-49.17	868.3	-2.30	785.6	97.25
	Flex-G2	210.5	-862.8	10.4	22.16	980.	2282
29	Fixed-H2	510.1	303.6	1449.	156.9	2304	-128.7
	Flex-H1	-83.3	-39.81	-225.5	32.36	947.5	14.05
	Flex-H3	-780.9	-235.4	-343.4	-34.3	-946.5	400.8
19	Fixed-A2	565.9	-35.48	810.0	0.1	1654.21	46.2
	Flex-A1	194.5	2.723	-1187.9	0.1	-1874.65	-6.6
	Flex-A3	-371.6	-222.62	43.4	3.3	56.94	411.6
14	Fixed-B3	-318.3	164.58	983.7	-13.2	1210.44	-193.8
	Flex-B1	172.5	37.47	-1089.9	0	-1494.46	-46.95
	Flex-B2	582.7	282.99	127.1	-0.119	218.98	-500.09
36	Fixed-C2	3055	635.8	3323	0.	4985	-2185
	Flex-C1	467.	60.98	-2107	0.	-2032	-75.3
	Flex-C3	-2438	-414.2	-574.	1.0	-656.4	559.3
41	Fixed-D2	290.7	29.4	-282.1	-9.3	-433.77	-31.42
	Flex-D1	76.4	1.92	380	-1.3	634.7	-4.11
	Flex-D3	-231.1	-71.8	27.4	-1.2	35.66	122.1

TABLE 2. STATIC LOAD SUMMARY

Load Condition: 17.8g x +3.3g z

Box No.	Attach Arm	P _X	V _Y	V _Z	M _X	M _Y	M _Z
		LBS	LBS	LBS	LB-IN.	LB-IN.	LB-IN.
25	Fixed-E2	3135.9	1017	1962	126.7	3043.2	-1676
	Flex-E1	-508	-156	-1154.9	10.62	-7.15	257.8
	Flex-E3	-2630.5	-312	-536.4	7.1	-1016.9	493.
16	Fixed-F2	908.2	-568	-2419.7	23.9	-3573.6	-13.7
	Flex-F1	431.	498.9	867.7	22.9	969.7	-964.7
	Flex-F3	-1381	-347	-391	39.4	-872.6	525.6
1	Fixed-G3	-404.	-341.	-2155.	-5.8	-2118.7	547.6
	Flex-G1	-44.7	78.4	475.5	3.3	300	-140.7
	Flex-G2	394	-850.6	88.6	33.0	2909.7	2325.
29	Fixed-H2	832.9	446.9	1312.6	143.9	2047.9	1326
	Flex-H1	-120	-57.1	-426	27.6	467	-20
	Flex-H3	-712	-171.7	-149.9	-24	-699	276
19	Fixed-A2	286.7	-329.4	803.3	-1.99	1104.9	551.6
	Flex-A1	141	45.2	-1181.8	0.184	-1836.8	-76.7
	Flex-A3	-446.9	-230.7	-36.2	1.41	-113.8	421.5
14	Fixed-B3	-503.4	-100.	977.	-13.9	1179.	178.2
	Flex-B1	107.9	32.36	-1078.7	0.	-1436	-35.9
	Flex-B2	395.5	300.5	304.4	0.05	476.2	-536.6
36	Fixed-C2	1950	-264.7	3082	0.	6397.7	1293
	Flex-C1	631.3	199.3	-2273.6	0.	-2332.9	-317.7
	Flex-C3	-2592	-485.7	-628.8	-.63	-639.1	660.6
41	Fixed-D2	314.7	104	-272.8	-9.5	-434	-151
	Flex-D1	-46.7	-9.5	392.6	-1.89	635.1	13.6
	Flex-D3	-267.9	-68.5	-22.4	-1.73	-19.8	115.1

TABLE 3. STATIC LOAD SUMMARY

Load Condition: 17.8g x -3.3g Y

Box No.	Attach Arm	P _X	V _Y	V _Z	M _X	M _Y	M _Z
		LBS	LBS	LBS	LB-IN.	LB-IN.	LB-IN.
25	Fixed-E2	2963.7	513	2046	113.5	3206	-836.4
	Flex-E1	324.6	-74	-1073	7.4	91.8	122.9
	Flex-E3	-2661	-311.5	-596.8	3.9	-1052.5	493.2
16	Fixed-F2	1999	-246	-2512	30.9	-3672	431.9
	Flex-F1	41.8	42.8	757.5	5.9	426.	-74.6
	Flex-F3	-1412	-372	-293	24	-241.5	582.
1	Fixed-G3	-307.	-180.9	-2090.	-12.5	-2264.	317.7
	Flex-G1	-13.2	72.3	509.7	3.2	494	-135.6
	Flex-G2	600.	-815.	250.8	30.5	3111.8	2223.5
29	Fixed-H2	792.	275.9	1426	117	2204	-117
	Flex-H1	203.5	-33.6	-285.8	15.8	629.7	6.2
	Flex-H3	-641	-198	-309	-26	-762	336.9
19	Fixed-A2	295.7	-67	757	-.008	1056	100
	Flex-A1	-207.2	7.01	-1230	1.0	-1899.7	-13.7
	Flex-A3	-523.6	-222.7	68.7	2.3	-29.6	404.6
14	Fixed-B3	-470.7	88.	916.8	-17.1	1013.7	-185
	Flex-B1	-169.7	-43.7	-1096.6	0.	-1320.6	102.96
	Flex-B2	203.5	343.	131.8	1.6	272.8	-631.
36	Fixed-C2	2244.5	633.2	3309.7	0.	6753	-156
	Flex-C1	-607	-5.3	-2018	0.	-1206	92.9
	Flex-C3	-2741	-510	-638.6	4.0	-720.7	691.8
41	Fixed-D2	240.1	41	-290	-7.1	-464	-501
	Flex-D1	-71.4	3.56	369.9	-1.4	592.8	-8.4
	Flex-D3	-304.7	-73.5	37.7	-1.3	92.6	122.3

TABLE 4. STATIC LOAD SUMMARY

Load Condition: 17.8g x -3.3g Z

Box No.	Attach Arm	P _X	V _Y	V _Z	M _X	M _Y	M _Z
		LBS	LBS	LBS	LB-IN.	LB-IN.	LB-IN.
25	Fixed-E2	2110	60.2	2143.9	103.8	3346.6	-88.8
	Flex-E1	509	-3.9	-946	2.6	212	6.7
	Flex-E3	-2644	-339	-701	.24	-1081	534.5
16	Fixed-F2	1679.8	39	-2551	38	-3801	950.9
	Flex-F1	-388.3	-423.2	681.6	-12.0	-184	835.8
	Flex-F3	-1347	-402.7	-212.7	7.1	529	633.5
1	Fixed-G3	-262.7	101.8	-1704.6	-14.4	-1806.	68.7
	Flex-G1	-203	-55	902.5	-2.4	979.6	102.4
	Flex-G2	416.	-827.	172.6	19.6	1182.	2180.
29	Fixed-H2	469.5	132.6	1563	130	2460	-378
	Flex-H1	240.6	-16.3	-85	20.5	1110	40.5
	Flex-H3	-710	-261.8	-503	-36.4	-1009	461
19	Fixed-A2	574.9	226.5	764.	2.1	1031	-404.8
	Flex-A1	-153.6	-35.5	-1236	1.0	-1937	56.2
	Flex-A3	-448	-214	148	4.2	141	394.7
14	Fixed-B3	-285.	352.	923.	-16.5	1045.	-557.
	Flex-B1	-105	-38.6	-1107	0.	-1379	91.9
	Flex-B2	390.7	325.4	-45.4	1.5	15.6	-594.
36	Fixed-C2	3349	1533	3550	0.	5397	-3766
	Flex-C1	-771	-143	-1852	0.	-906	335
	Flex-C3	-2587	-438.9	-584.6	5.6	-738	590
41	Fixed-D2	216	-33	-299.7	-6.95	-463.7	69.9
	Flex-D1	51.7	14.9	357	-.97	592	-26
	Flex-D3	-267.9	-76.8	87.7	-.8	148	129

TABLE 5. DYNAMIC FLEX ARM LOADS

Panel Base Input: 3.85g, Panel Frequency: 39 Hz

Box No.	Attach Arm	P _X	V _Y	V _Z	M _X	M _Y	M _Z
		LBS	LBS	LBS	LB-IN.	LB-IN.	LB-IN.
25	Fixed-E2	2625.2	3728.5	-1712.2	48.0	-1657.1	-6196.1
	Flex-E1	-4445.3	-594.0	-113.9	68.8	-632.0	978.9
	Flex-E3	806.3	442.1	319.1	65.0	44.3	-694.0
16	Fixed-F2						
	Flex-F1						
	Flex-F3						
1	Fixed-G3						
	Flex-G1						
	Flex-G2						
29	Fixed-H2	-1073.4	-783.2	45.4	-47.8	10.4	1249.9
	Flex-H1	973.6	114.7	558.1	-16.1	763.6	-186.2
	Flex-H3	838.6	-9.3	271.3	-11.6	237.0	-351.3
19	Fixed-A2	741.2	-282.9	-459.5	-14.9	-662.7	561.1
	Flex-A1	360.9	56.2	977.1	-4.2	1686.5	-89.6
	Flex-A3	-142.0	139.0	388.0	-6.5	447.6	-249.7
14	Fixed-B3	-1076.9	1196.3	307.9	-25.5	721.2	-1955.1
	Flex-B1	-405.9	-282.9	1331.6	0.	2101.3	457.6
	Flex-B2	1126.1	-13.9	769.7	-13.0	1160.4	53.1
36	Fixed-C2	-225.1	1394.3	-1332.4	0.	-1311.2	-2173.4
	Flex-C1	-1956.3	-405.6	1830.	0.	2990.4	761.0
	Flex-C3	1305.4	278.7	264.5	4.9	174.3	-376.8
41	Fixed-D2	155.6	158.2	82.7	-1.2	123.9	-234.8
	Flex-D1	-238.7	-22.7	-44.6	-1.4	-79.6	37.6
	Flex-D3	-15.8	30.4	-38.8	-1.2	-57.0	-51.2

TABLE 6. DYNAMIC FLEX ARM LOADS

Panel Base Input: 0.37g Y, Panel Frequency: 10.38 Hz

Box No.	Attach Arm	P _X	V _Y	V _Z	M _X	M _Y	M _Z
		LBS	LBS	LBS	LB-IN.	LB-IN.	LB-IN.
25	Fixed-E2	699.45	-535	288	-6.66	576.61	895.3
	Flex-E1	1558.5	87.4	-12.73	-0.16	145.9	-144.2
	Flex-E3	-373.52	0.87	-119.4	-1.33	-174.6	2.17
16	Fixed-F2						
	Flex-F1						
	Flex-F3						
1	Fixed-G3						
	Flex-G1						
	Flex-G2						
29	Fixed-H2	201.	-353.	121.	1.25	184	544
	Flex-H1	369	45.5	28	1.17	119.8	-74.7
	Flex-H3	-238.4	2.37	-172.7	-2.37	-129.6	10.00
19	Fixed-A2	-646.7	457.3	-21.4	1.54	-231.4	-771.9
	Flex-A1	-702.2	-63.5	55.5	1.48	136.	105.6
	Flex-A3	204.3	37.11	229.2	0.85	150.4	-77
14	Fixed-B3	842.1	373.2	-139.75	2.96	-211.1	-580.9
	Flex-B1	-137.7	-84.9	121.5	0.	293.8	143.8
	Flex-B2	-843.9	-23.9	-316.7	1.96	-517.1	23.7
36	Fixed-C2	-108.3	1449.8	-185.4	0.	-204	-2283
	Flex-C1	-2263.6	-336.1	724.5	0.	2277	685.6
	Flex-C3	217.15	15.02	130.3	8.7	-50.2	-23.8
41	Fixed-D2	-504	-165	-3.74	4.51	-9.2	272
	Flex-D1	-44.4	33	-86.1	0.71	-130.4	-54
	Flex-D3	-50	-4.2	204.5	0.78	349	5.29

TABLE 7. DYNAMIC FLEX ARM LOADS

Panel Base Input: 0.52g Z, Panel Frequency: 9.82 Hz

Box No.	Attach Arm	P_X	V_Y	V_Z	M_X	M_Y	M_Z
		LBS	LBS	LBS	LB-IN.	LB-IN.	LB-IN.
25	Fixed-E2	2531.4	1370.7	-185.9	32.3	-214.6	-2271.
	Flex-E1	-737.2	-217.3	-366.6	15.1	-327.0	356.9
	Flex-E3	-184.2	58.4	221.2	12.0	18.7	-91.0
16	Fixed-F2						
	Flex-F1						
	Flex-F3						
1	Fixed-G3						
	Flex-G1						
	Flex-G2						
29	Fixed-H2	1209.	801.3	135.7	7.5	189.7	-1276.
	Flex-H1	-320.7	-118.4	-55.6	2.6	-108.2	191.9
	Flex-H3	-341.1	16.1	142.7	-.05	122.6	-30.4
19	Fixed-A2	-1334.7	-918.8	-105.8	-7.6	-62.4	1599.5
	Flex-A1	247.8	143.5	287.6	-1.0	505.9	-234.5
	Flex-A3	286.8	4.8	-313.7	-6.1	-459.1	-20.6
14	Fixed-B3	441.2	-732.7	35.6	5.3	162.4	1216.7
	Flex-B1	285.3	87.0	304.7	0.	263.6	-157.3
	Flex-B2	-806	-63.9	528.	-2.5	754.	126.7
36	Fixed-C2	-3893.	-3134.	-1577.1	0.	-4072.6	6339.8
	Flex-C1	1146.1	496.9	-126.3	0.	-1093.	-909.5
	Flex-C3	440.4	-89.8	20.0	-8.0	260.3	133.
41	Fixed-D2	-272.	377.7	50.9	-0.4	36.0	-605.8
	Flex-D1	-260.2	-656.8	25.2	-1.2	-3.2	107.8
	Flex-D3	8.5	14.6	-336.2	-1.4	-493.5	-28.4

TABLE 8. NOTCHED INPUT AT THE BASE OF THE SPACECRAFT AND PEAK RESPONSE VALUES AT TOP CORNER OF PANEL

Notched Input G 112 - X Axis, Response G 2001 - 1X

<u>Frequency</u> <u>Hz</u>	<u>Acceleration</u> <u>g</u>	<u>Peak Responses</u> <u>g</u>
62.3	.58	13.1

Notched Input G 112 - Y Axis, Response G 2001 - 2Y

<u>Frequency</u> <u>Hz</u>	<u>Acceleration</u> <u>g</u>	<u>Peak Responses</u> <u>g</u>
8.20	0.59	10.23
9.12	0.57	10.34
18.70	0.75	10.45
21.41	0.75	3.63
33.8	0.75	1.80
47.7	0.75	7.80
74.57	0.75	1.50

Notched Input G 112 - Z Axis, Response G 2001 - 3Z

<u>Frequency</u> <u>Hz</u>	<u>Acceleration</u> <u>g</u>	<u>Peak Responses</u> <u>g</u>
8.20	0.55	7.92
9.12	0.54	7.34
28.2	0.75	1.75
33.8	0.75	0.63
41.0	0.75	0.65
62.38	0.75	1.05
74.57	0.75	0.45

SECTION 3

DESCRIPTION OF THE MODEL

3.1 Description of the structure. The structure consists of an instrument support panel (ISP) held by three support structures--the fixed support (Figure 3), the pivot link support (Figure 2), and the side support (Figure 4). All three supports have ball joints with the panel. The instrument support panel, made of aluminum, has an eggcrate structure and supports eight simulated instrument boxes, four on each side, by means of two flex arms and one fixed arm for each of the boxes. One edge of the panel is held by a fixed support while the other edge is held by a pivot link support. Also, one side of the panel is held by a strut connected to the base structure through a pedestal.

3.2 The finite element model.

3.2.1 Instrument support panel. The panel is made in an eggcrate configuration and is modeled as 540 triangular plates (Figure 1). An equivalent core thickness and shear modulus are calculated as shown in Section 2.

The panel is supported at three places. Doubler plates are provided at the three places for interfacing with the support structures. The doubler plates at the fixed and pivot support (Figure 5) are added on as extra wall thickness to the corresponding triangular plates in that region. However, the doubler plate at the side support (Figure 6) is treated as a different set of quad plates and attached to the panel by means of fictitious rigid beams. The panel supports the light instrument boxes by means of 16 flex arms and 8 fixed arms. Each flex arm/fixed arm is attached to the panel at the appropriate location by a set of four rigid beams connecting it to the four nearest nodes in the panel.

3.2.2 Instrument boxes. The instrument boxes (Figures 7 through 14) are modeled as a combination of quad plates, triangular plates, and beams, according to the drawings supplied by NASA. These are simulated boxes to be used in dynamic test. Some small sections of the boxes are modeled as beams with equivalent mass points. Each box is attached to the panel by means of two flex arms and one fixed arm.

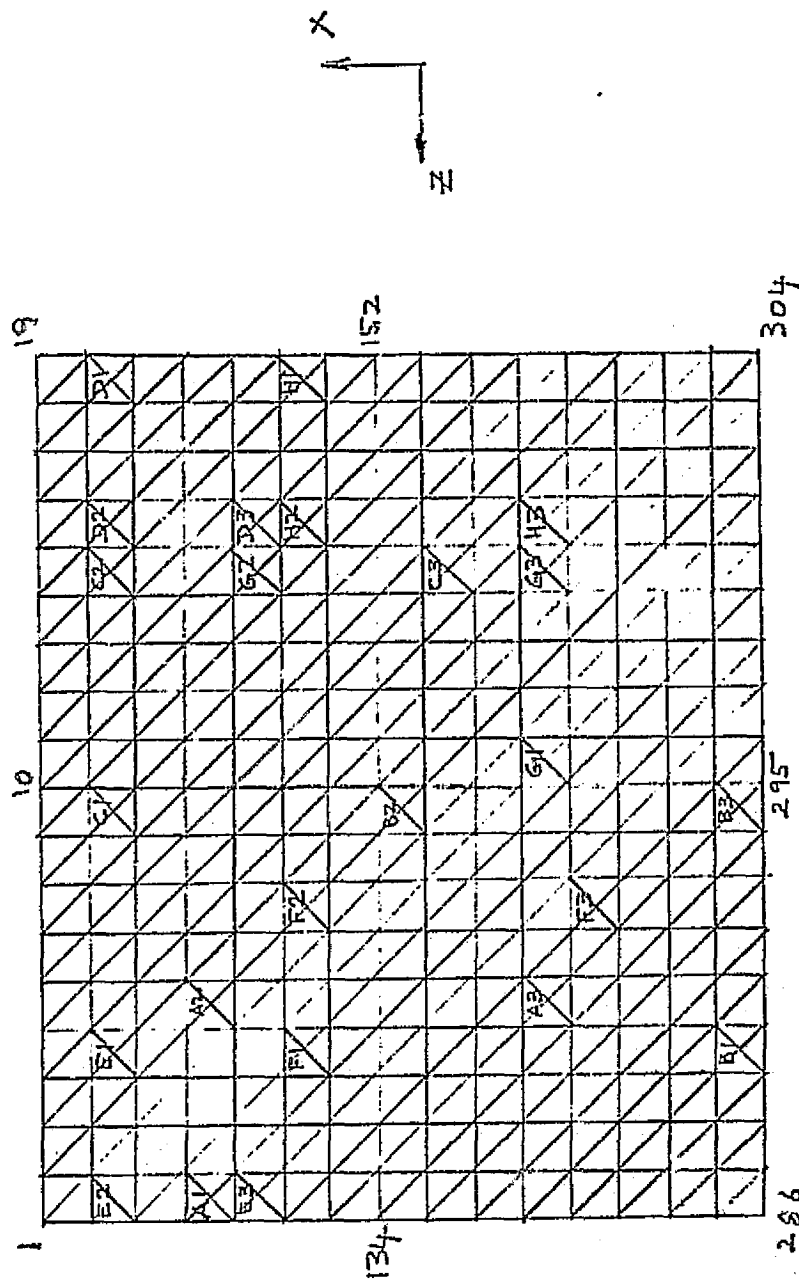
3.2.3 Platform structure. Since the bases of the fixed and pivot supports were completely restrained, it was required to model only that part of the platform structure which is directly under the pedestal side support (Figure 4).

3.2.4 The side strut. The strut providing the side support has ball joints at both ends. Hence, it is modeled as an axial element.

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INSTRUMENT SUPPORT PANEL

FIGURE 1

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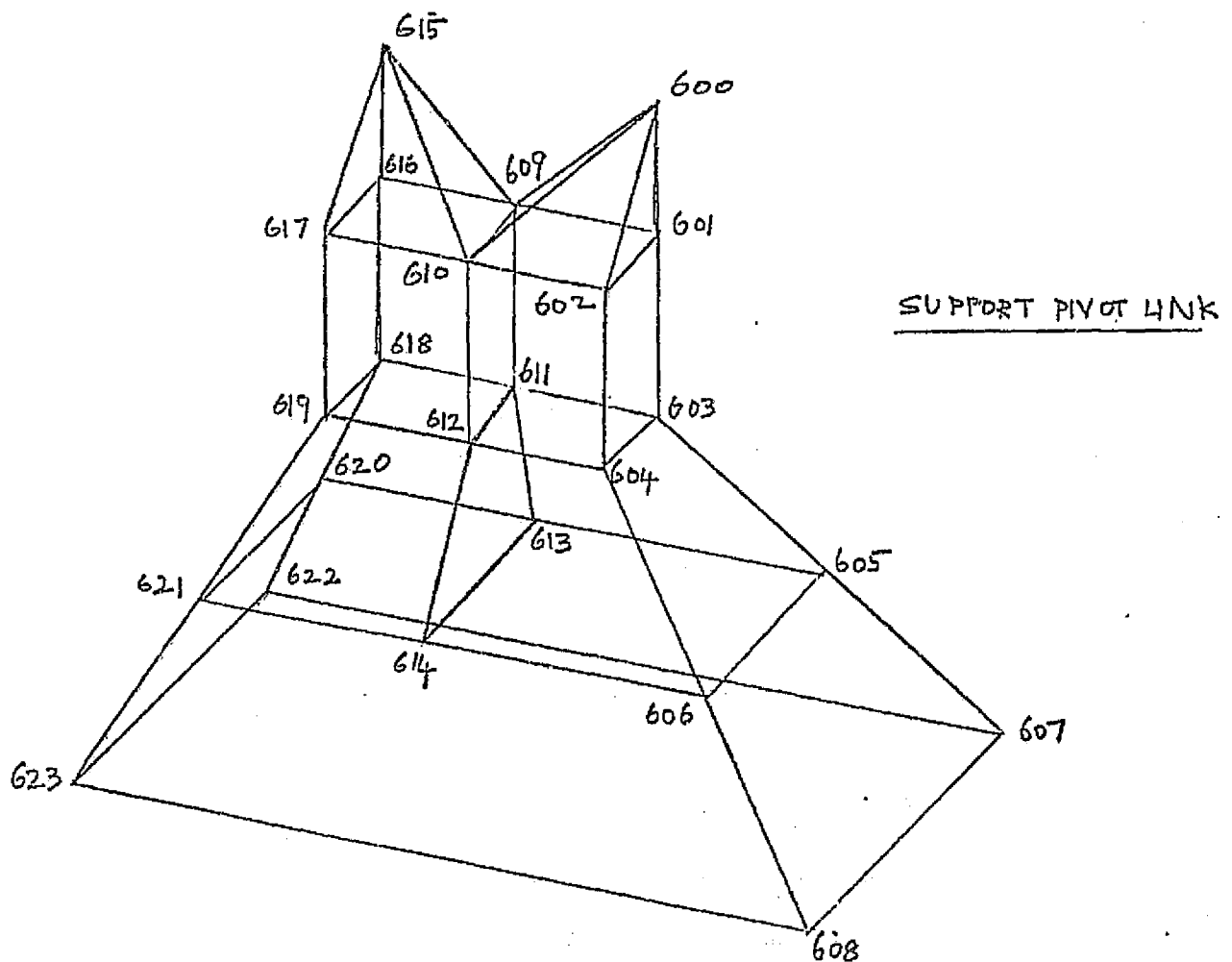
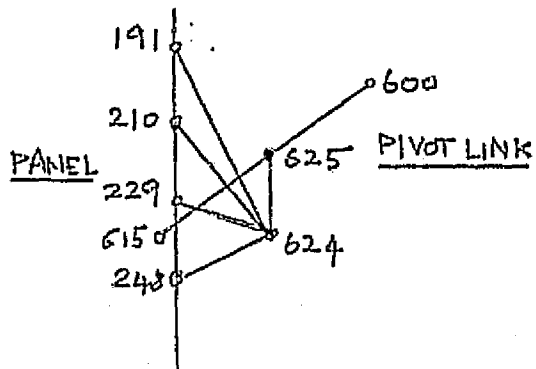


FIGURE 2

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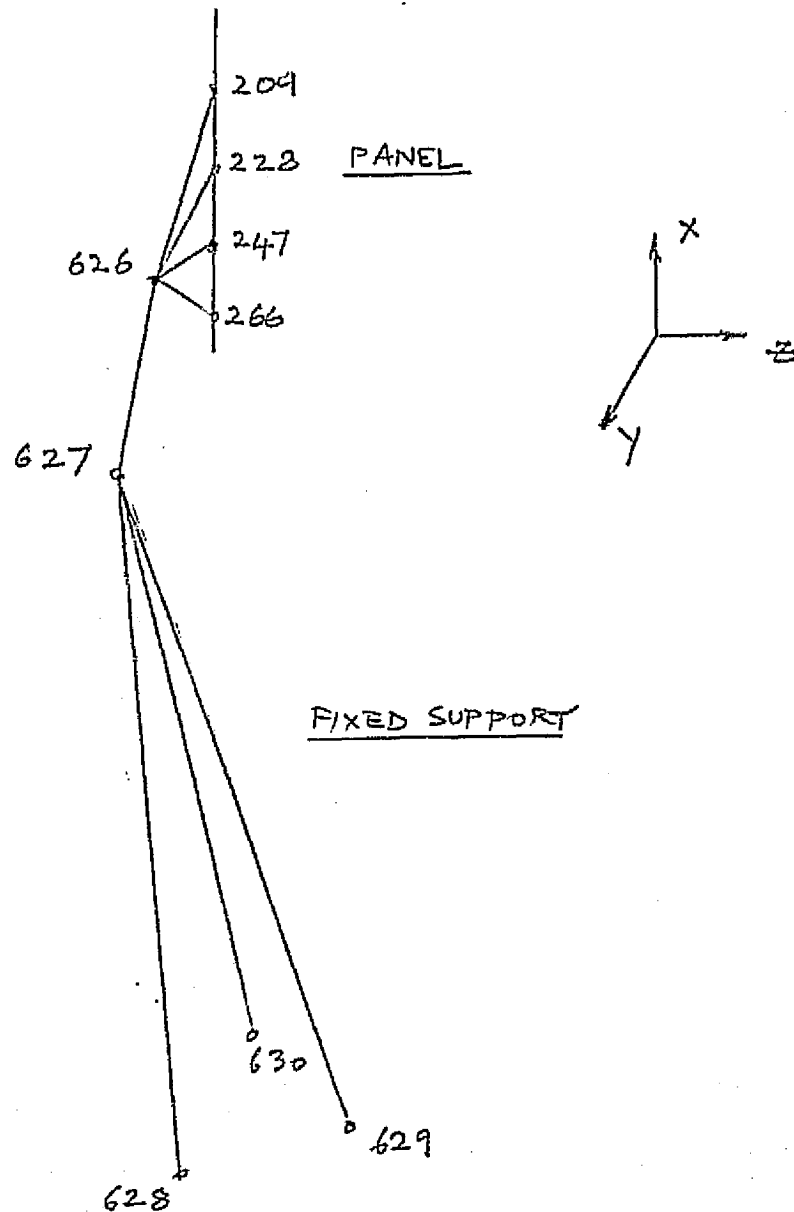


FIGURE 3

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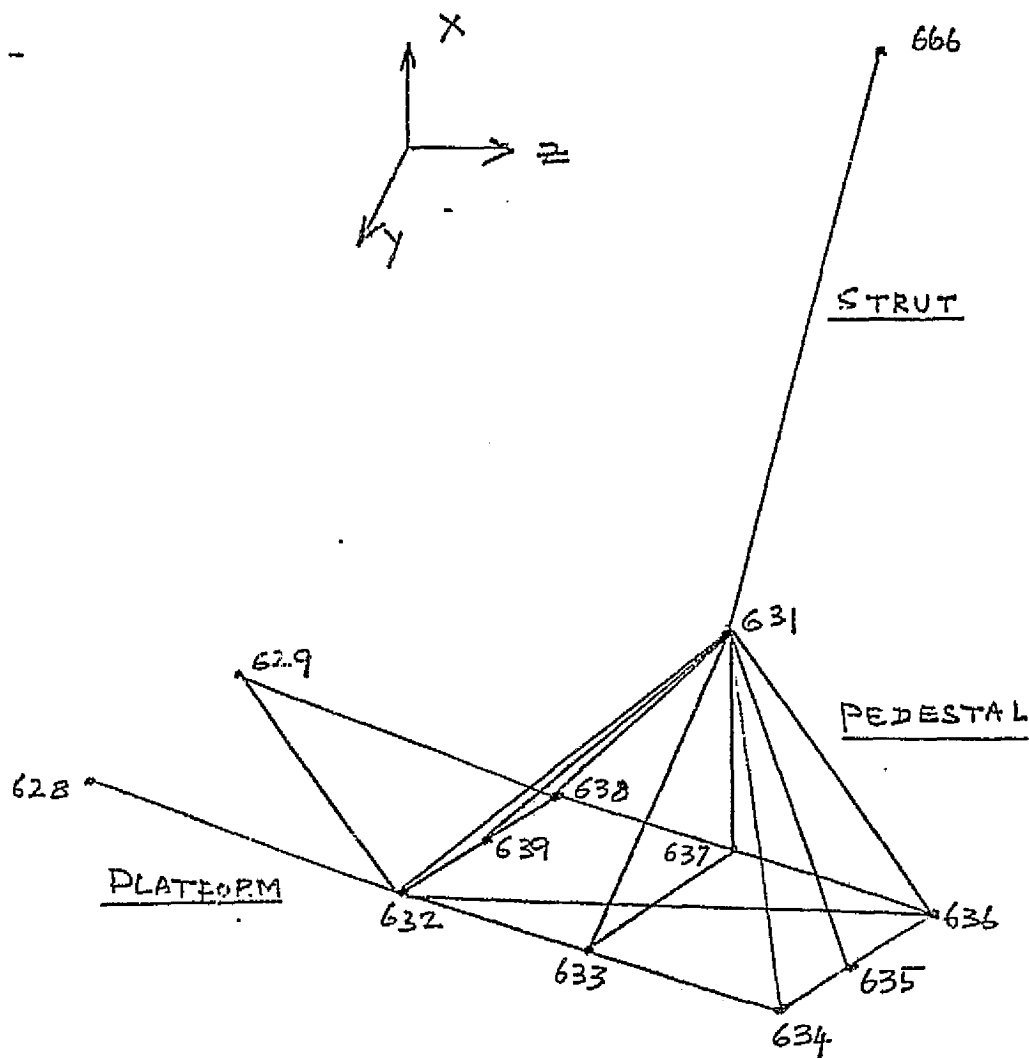


FIGURE 4

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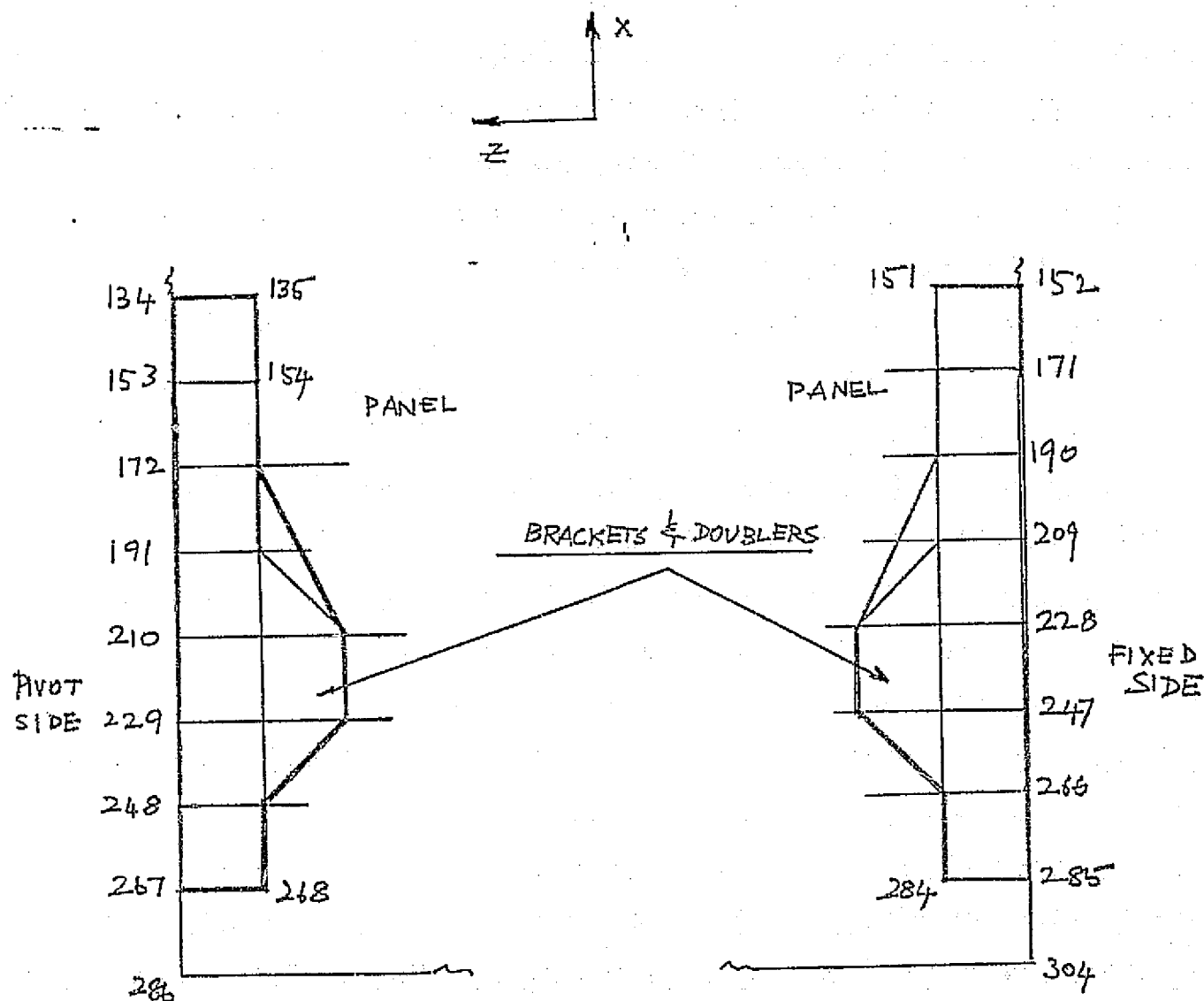


FIGURE 5

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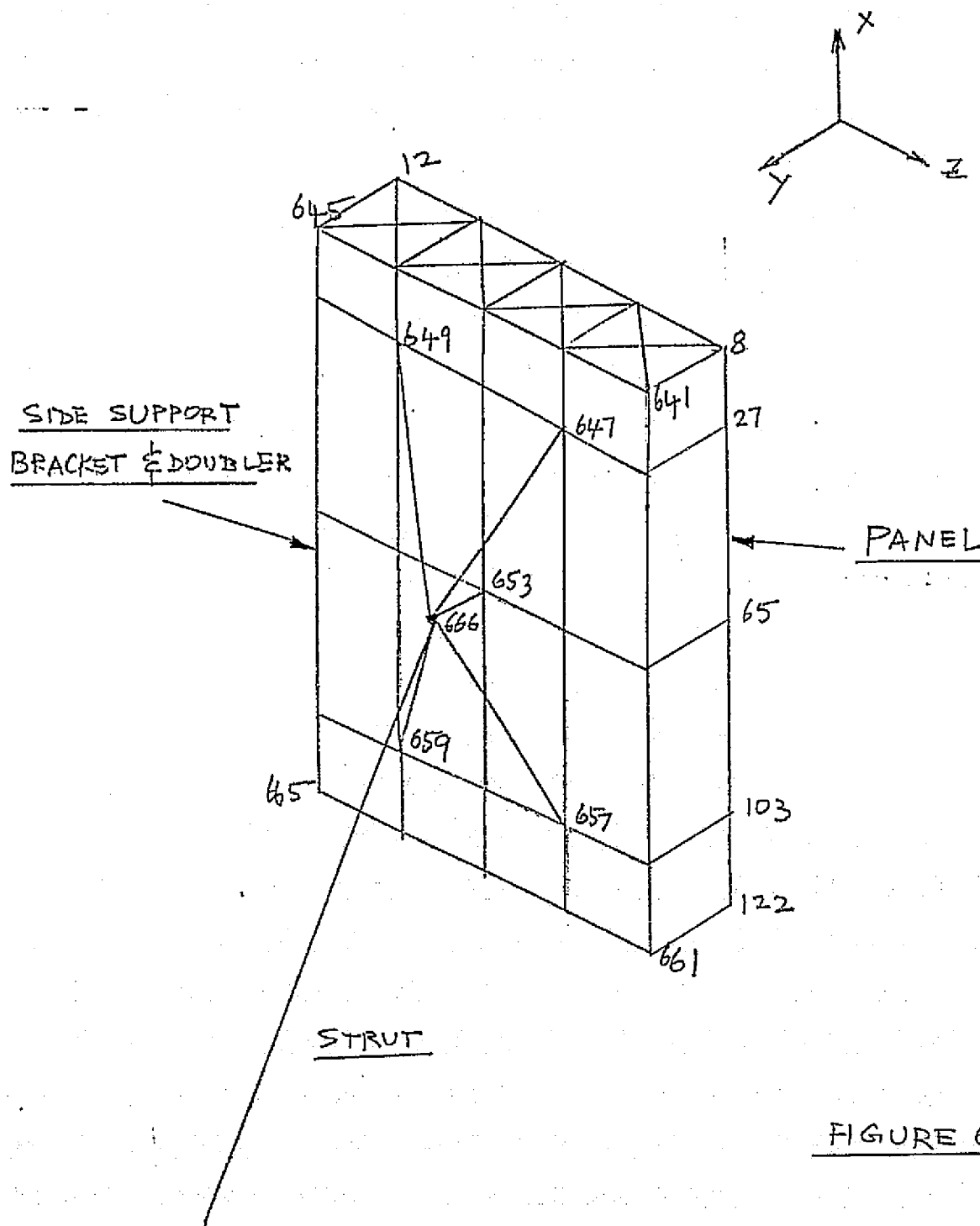
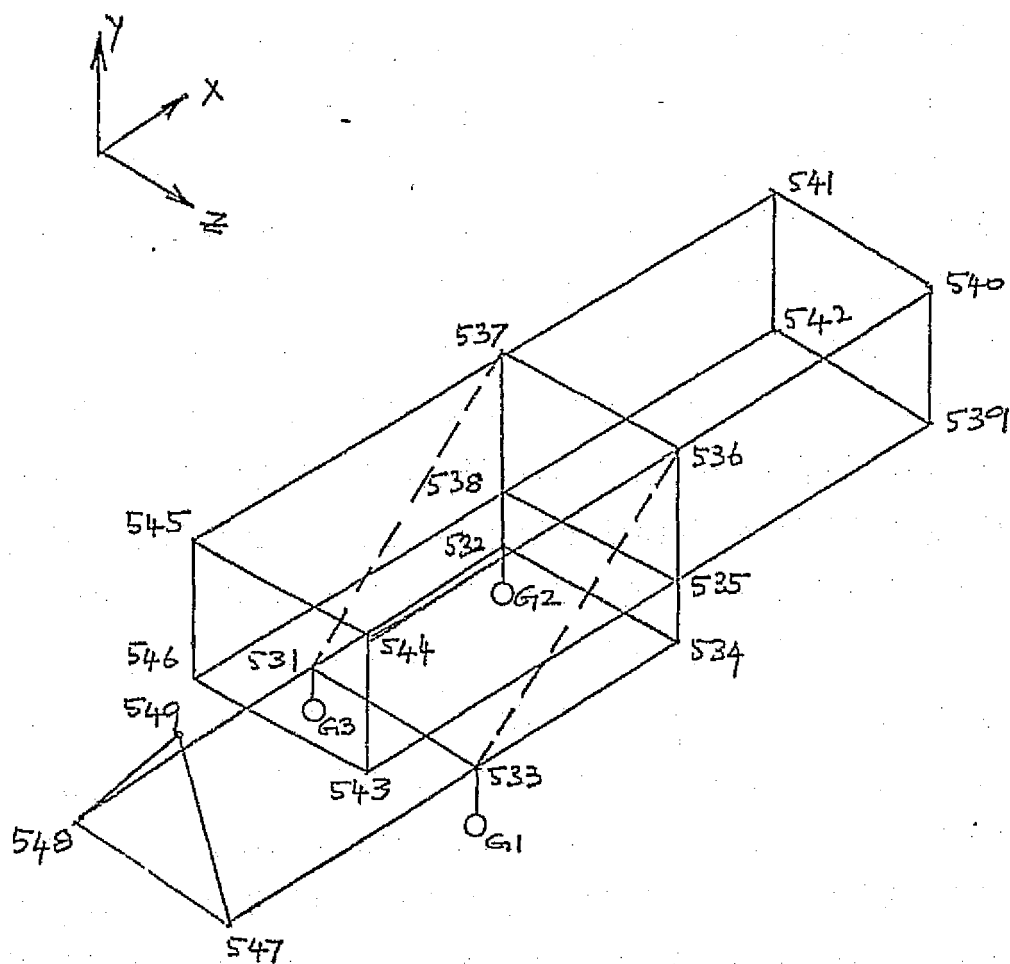


FIGURE 6

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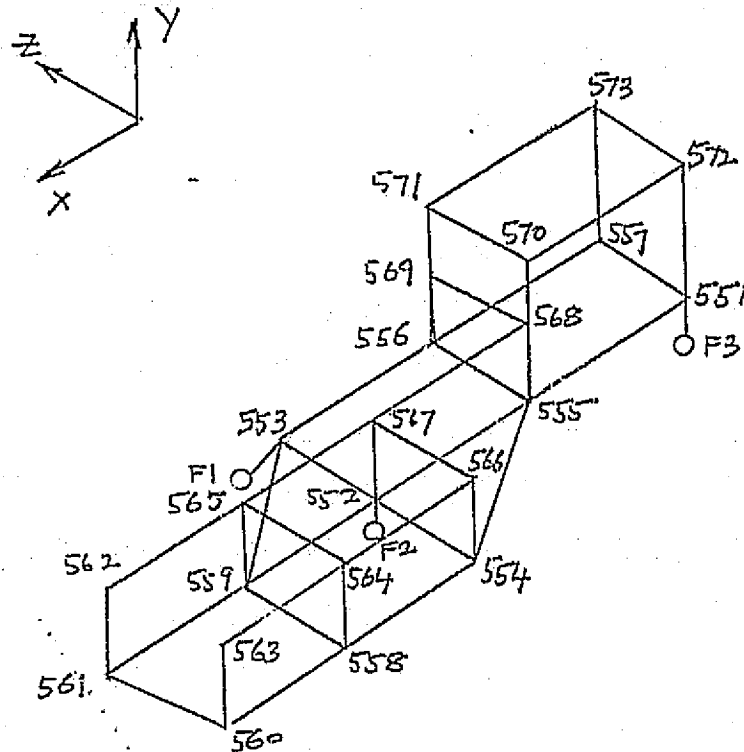
TWO BEAMS ADDED BETWEEN
531 & 537 AND 533 & 536 FOR
DYNAMIC ANALYSIS.

INST. # 1 WT. 86.90 LB

FIGURE 7

ASTRO
ASTRO Electronics
Division
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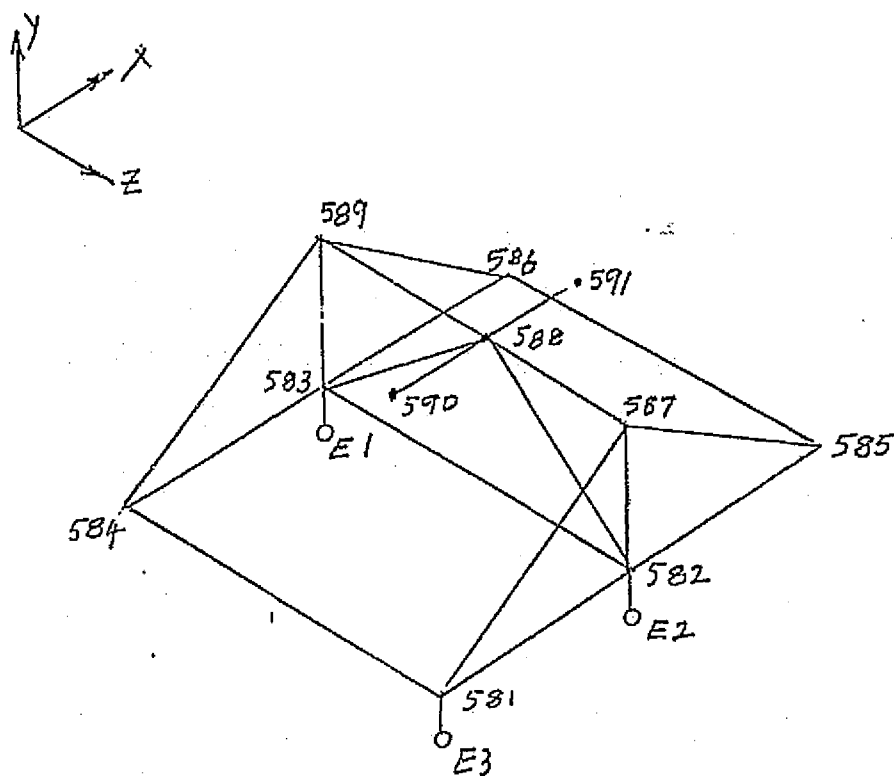
INST. # 16

WT. 203.81 LB

FIGURE 8

ASTRO
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Division
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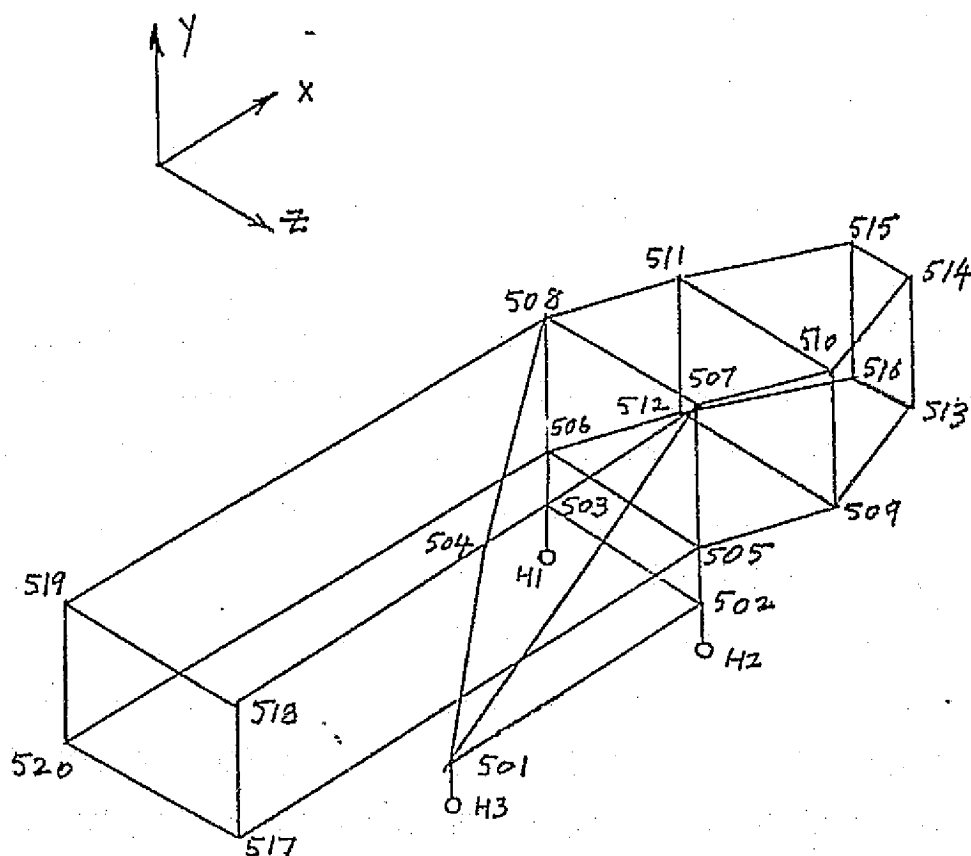
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INST. # 25 WT. 135.54B
FIGURE 9

R. C. W. E.
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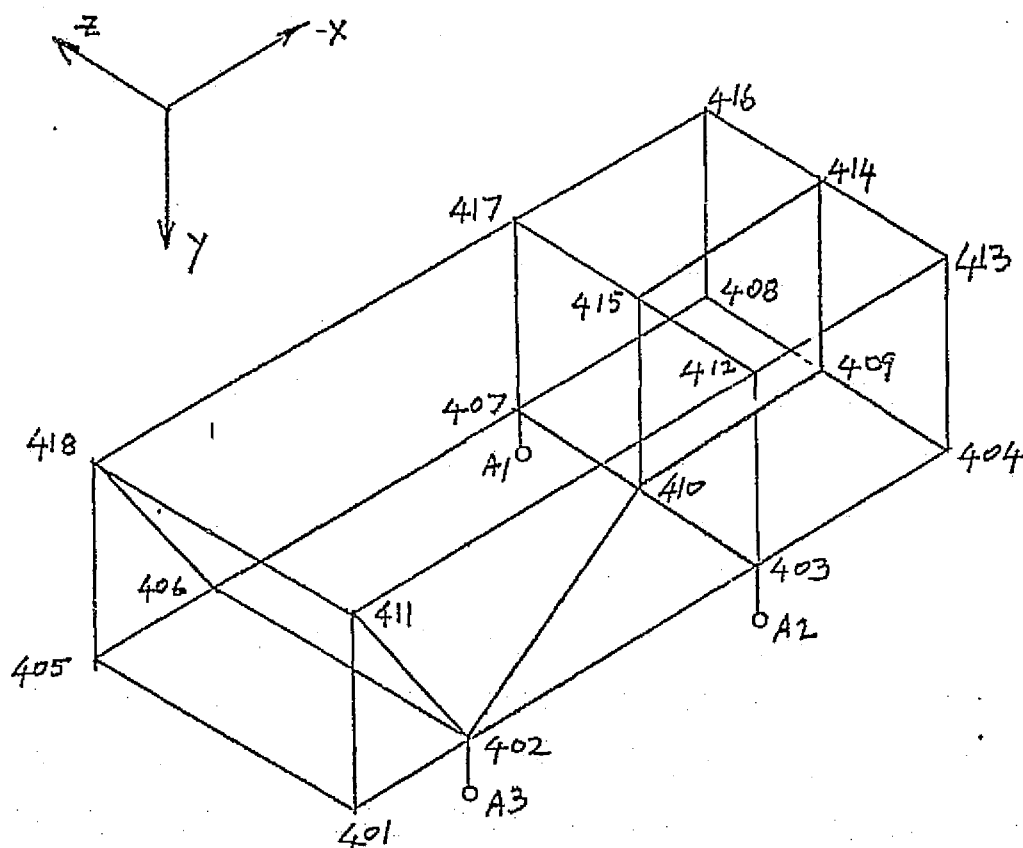
INST. # 29
WT. 94.78 LB

TWO BEAMS ADDED BETWEEN 502 & 505
AND 503 & 506 FOR DYNAMIC ANALYSIS

FIGURE 10

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INST. # 19

WT. 112.16 LB

FIGURE 11

ECG/71

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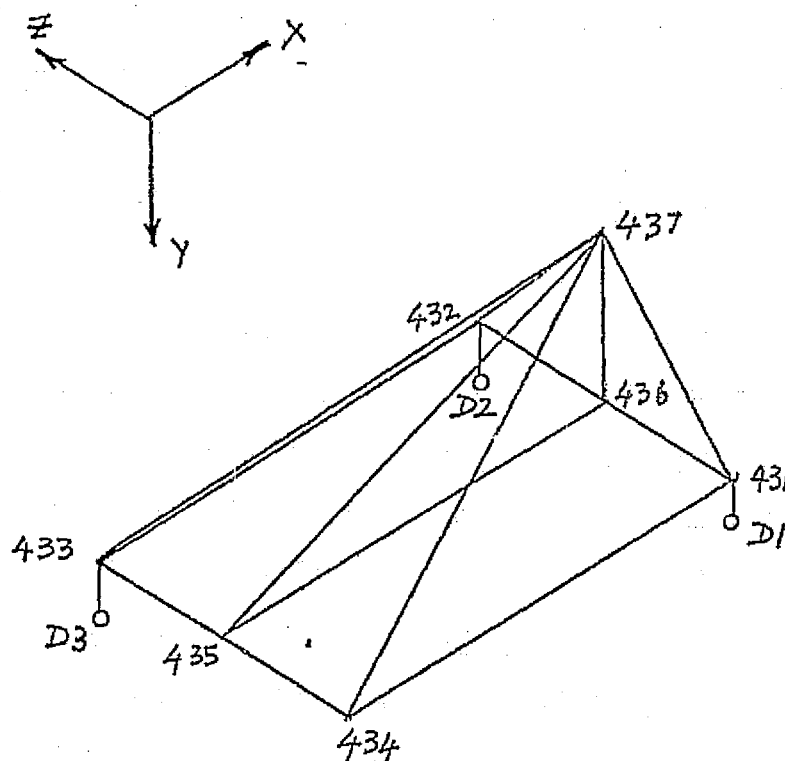
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INST. # 41 WT. 41.02 LB

FIGURE 12

U. G. M.

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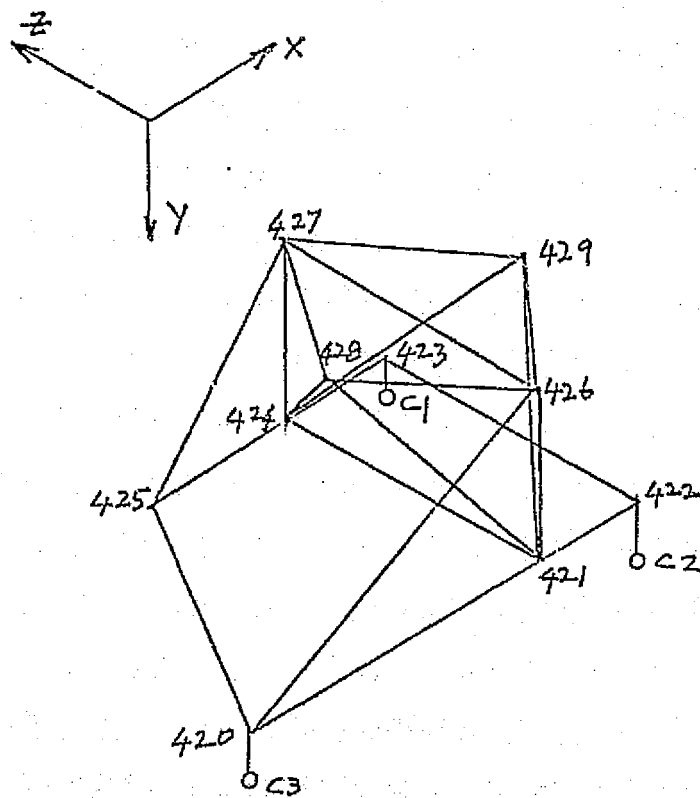
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Subject: INSTRUMENT NO. 36 Page No. ____ of ____



INST. # 36

WT. 331.0 LB

FIGURE 13

ENCLOSURE

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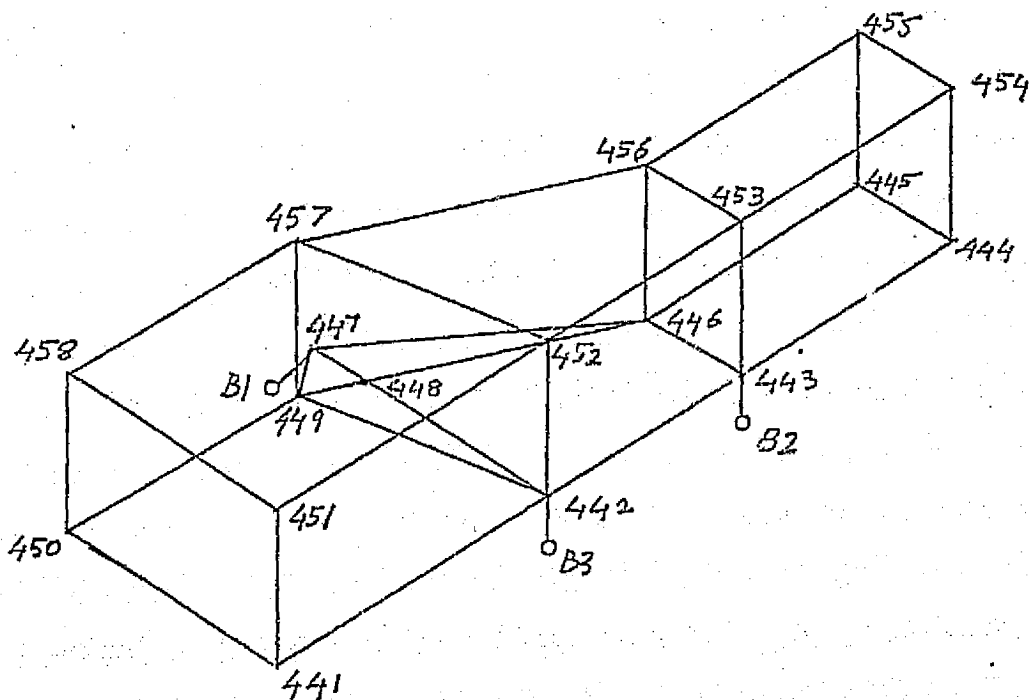
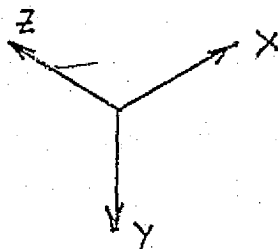
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INST. # 14 WT. 132.2 LB

FIGURE 14

SECTION 4

EQUIVALENT PANEL PROPERTIES

RODA

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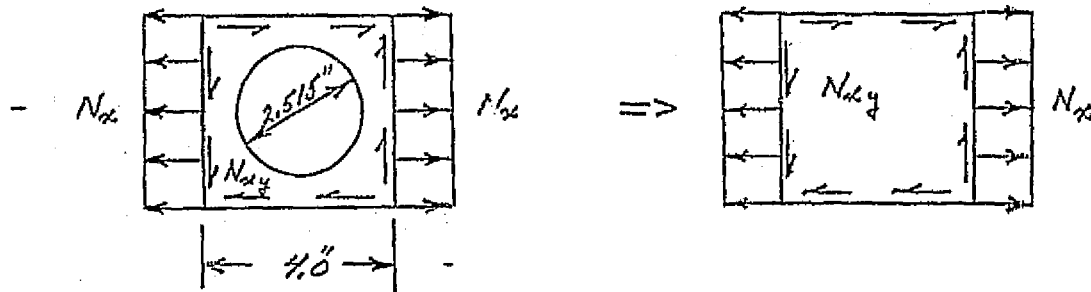
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EQUIVALENT PANEL PROPERTIES



PURPOSE: DETERMINE AN EQUIVALENT THICKNESS FOR THE PANEL FACE SHEETS WITH EQUAL AXIAL AND SHEAR STIFFNESS.

ASSUMING EQUIVALENT MATERIAL PROPERTIES (E, G, ν) THIS RESULTS IN AN EQUIVALENT THICKNESS, t_e , FOR THE FACE SHEETS.

METHOD: THE EQUIVALENT PLATE SHOULD HAVE THE SAME STRAIN ENERGY AS THE ACTUAL PLATE FOR AN EQUAL LOADING.

ROBAM

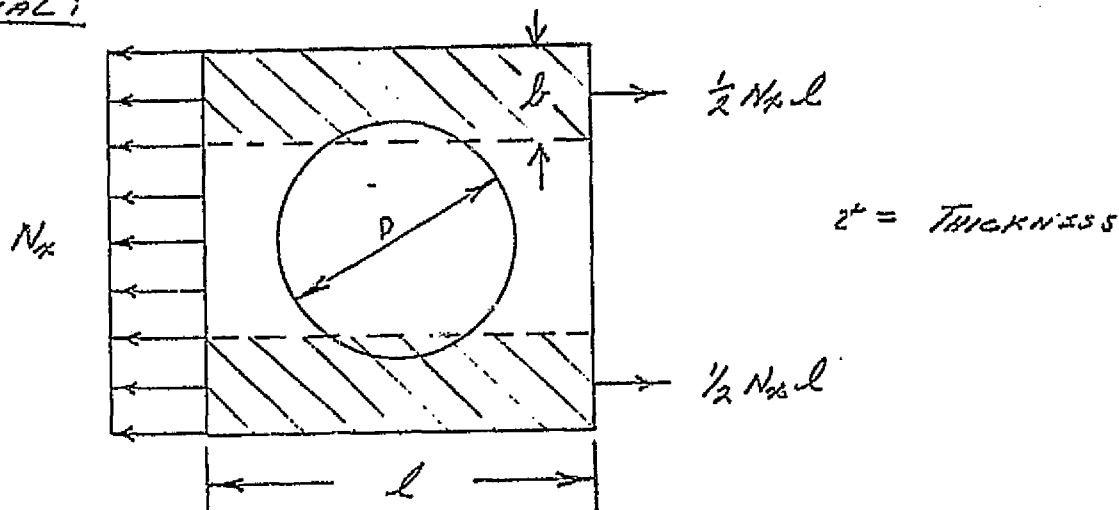
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STRAIN ENERGY IN FACE SHEET:

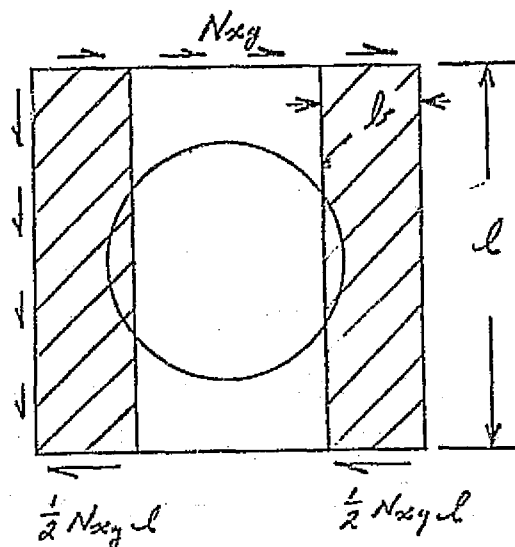
EQUIVALENT AREA IN ACTUAL FACE SHEET:

AXIAL:



$$\text{STRAIN ENERGY: } U = 2 \left[\frac{1}{2} \left(\frac{1}{2} N_x l \right)^2 l \left(\frac{1}{t E} \right) \right] = \frac{N_x^2 l^3}{4 t E}$$

SHEAR:



$$\text{STRAIN ENERGY: } U = 2 \left[\frac{1}{2} \left(\frac{1}{2} N_{xy} l \right)^2 l \left(\frac{1}{t G} \right) \right] = \frac{N_{xy}^2 l^3}{4 t G}$$

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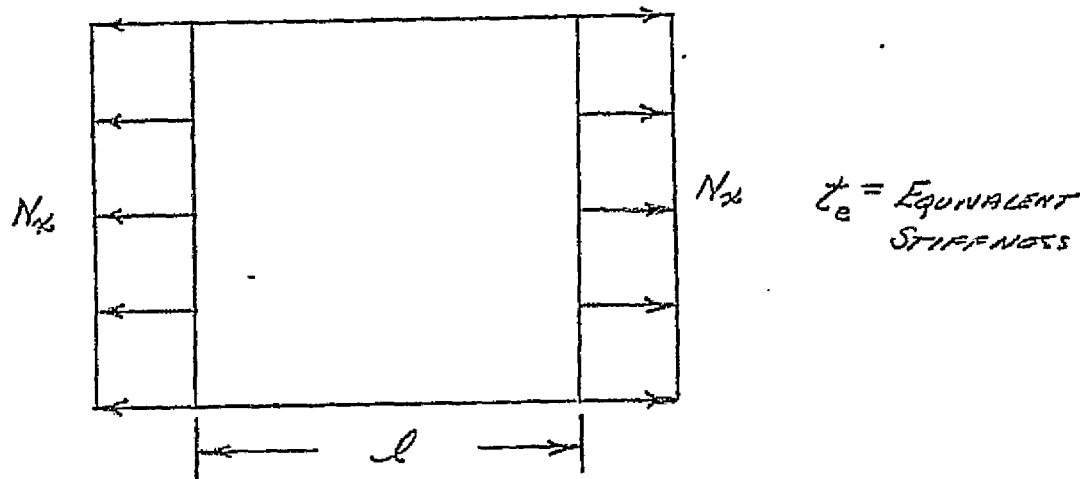
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STRAIN ENERGY IN EQUIVALENT FACE SHEET:

PRIAL:



STRAIN ENERGY:
$$U_e = \frac{1}{2} \frac{N_x^2 l^3}{l t_e E} = \frac{N_x^2 l^2}{2 t_e E}$$

EQUATING THE STRAIN ENERGY IN THE ACTUAL AND EQUIVALENT PLATE: $U \equiv U_e$

$$\frac{N_x^2 l^3}{4 l t_e E} = \frac{N_x^2 l^2}{2 t_e E}$$

OR

$$t_e = \left(\frac{2b}{l} \right) t$$

THE SAME RESULT IS OBTAINED FOR AN EQUIVALENT SHEAR STRAIN ENERGY IN THE FACE SHEET.

PRODA

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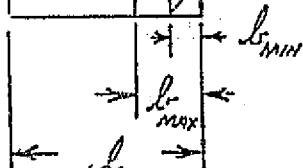
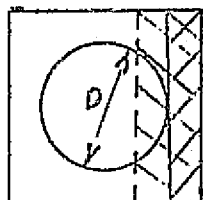
EQUIVALENT FACE SHEET THICKNESS:

NUMERICAL DATA:

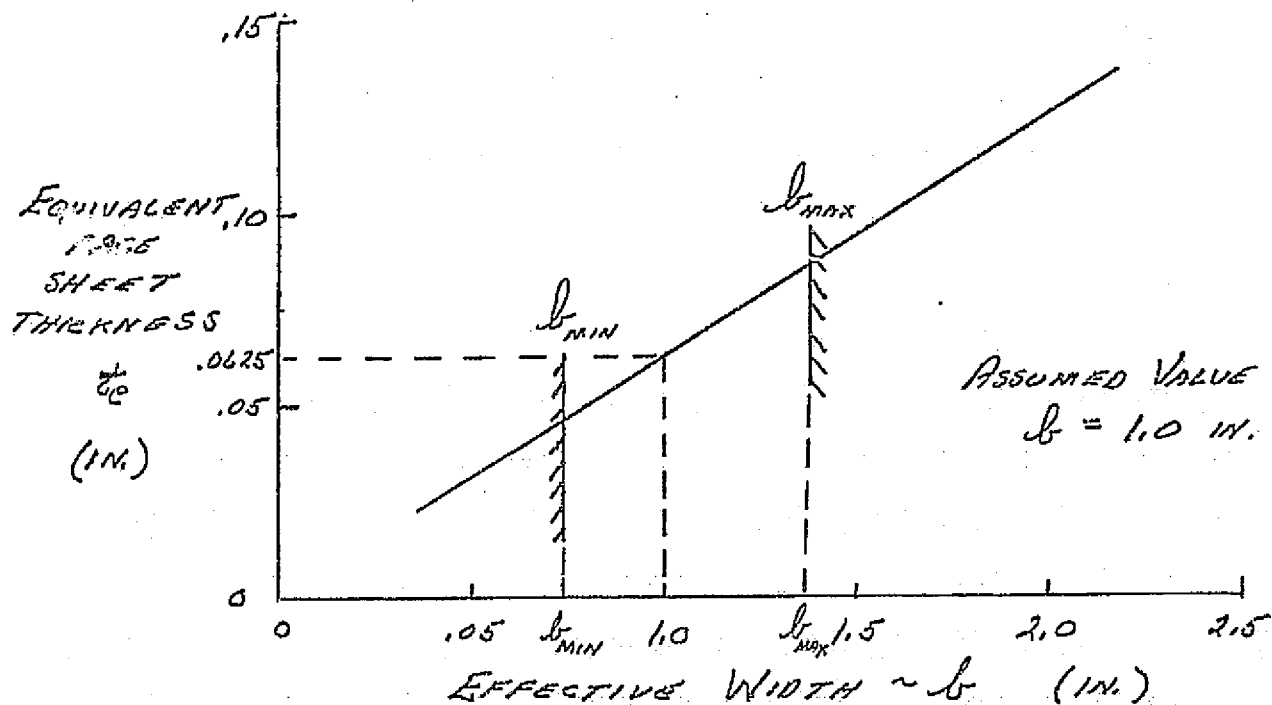
$$L = 4.0" \quad \frac{1}{2}(L-D) \leq b \leq \frac{1}{2}(L + \frac{D}{2})$$

$$D = 2.515" \quad \text{THEREFORE:}$$

$$t = .125" \quad (.74 \leq b \leq 1.37)$$



$$t_e = \left(\frac{2b}{L} \right) t = \left(\frac{2b}{4} \right) (.125) = .0625 b$$



THEREFORE:

$$t_e = .0625 \text{ IN.}$$

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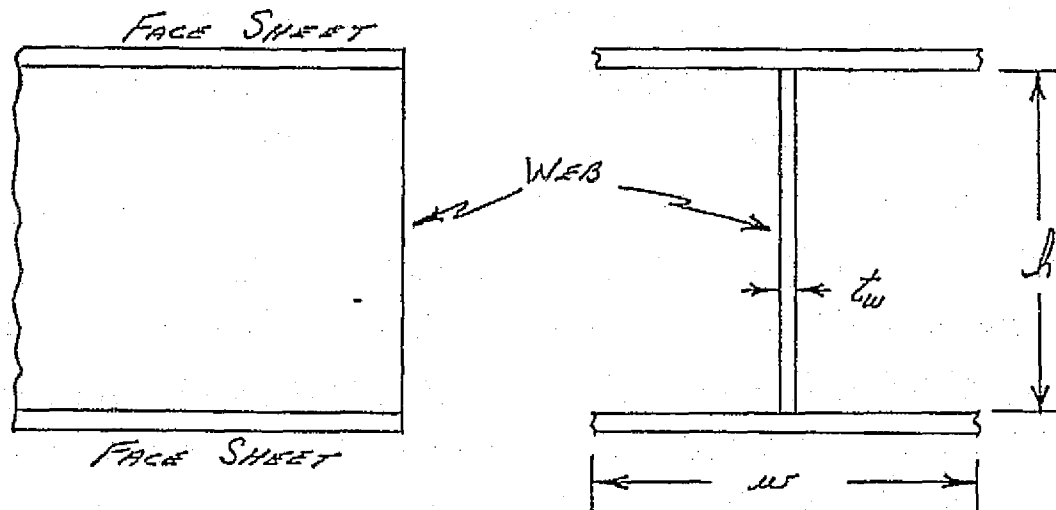
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Approved: _____ Project: S.M.M.

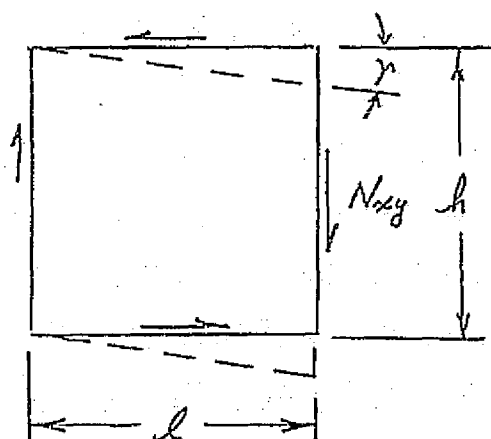
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EQUIVALENT CORE SHEAR MODULUS:



TRANSVERSE SHEAR STIFFNESS:



t_w = THICKNESS OF WEB

$$\gamma = \frac{N_{xy} h l}{t_w h G} = \frac{N_{xy} l}{t_w G}$$

EQUIVALENT ISOTROPIC CORE SHEAR MODULUS: $\gamma \equiv \gamma_e$

WHERE
$$\gamma_e = \frac{N_{xy} h l}{w h G_e} = \frac{N_{xy} l}{w G_e}$$

THEREFORE:

$$G_e = \left(\frac{t_w}{w} \right) G$$

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EQUIVALENT CORE SHEAR MODULUS (CON'T)

NUMERICAL DATA:

$$G = 3.8 \times 10^6 \text{ PSI} \quad (6061-T4 \text{ \& -T6})$$

$$t_w = .063 \text{ IN.}$$

$$w = 4.0 \text{ IN}$$

THEREFORE:

$$G_e = \left(\frac{t_w}{w} \right) G$$

$$= \left(\frac{.063}{4.0} \right) (3.8 \times 10^6)$$

$$G_e = .060 \times 10^6 \text{ PSI}$$

1003/71

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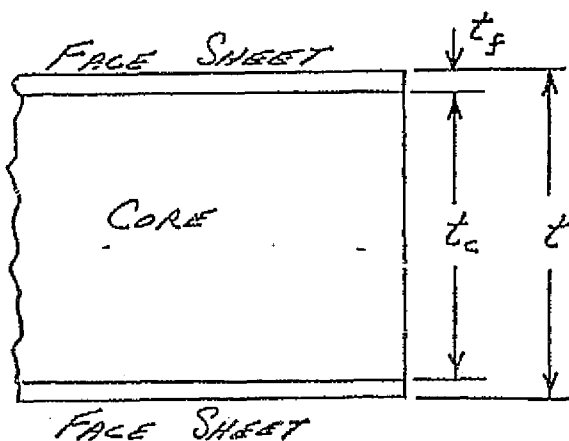
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SUMMARY OF PANEL PROPERTIES:



EQUIVALENT PANEL PROPERTIES:

$$t_f = .0625 \text{ IN}$$

$$t_c = 4.0 \text{ IN}$$

$$t = 4.125 \text{ IN}$$

MATERIAL PROPERTIES:

FACE SHEETS:

$$E = 10.0 \times 10^6 \text{ PSI} \quad (6061-T4)$$

$$G = 3.8 \times 10^6 \text{ PSI}$$

$$\alpha = 13.0 \times 10^{-6} \text{ IN/IN/}^\circ\text{F}$$

EQUIVALENT CORE:

$$G_c = .060 \times 10^6 \text{ PSI}$$

SECTION 5

BEAM SECTION PROPERTIES

INCOB/A

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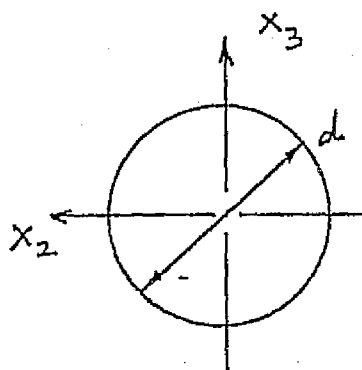
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$$A = \pi d^2/4$$

$$I_2 = I_3 = \pi r^4/4$$

$$J = I_2 + I_3 = \pi r^4/2$$

$$SF2 = SF3 = 0.89$$

BPROP # 1

$$d = 1.5''$$

$$A = \pi (1.5)^2/4 = 1.767$$

$$I_2 = I_3 = \pi (0.75)^4/4 = 0.249$$

$$J = 2(0.249) = 0.497$$

TR-0371

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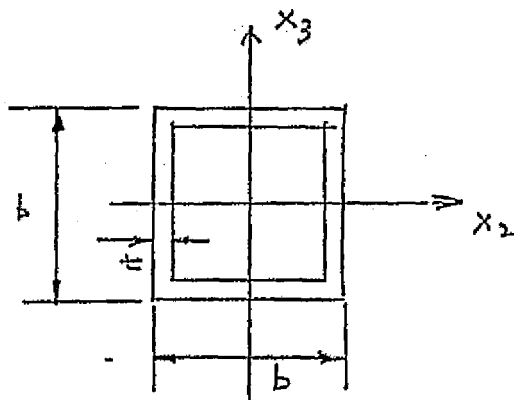
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$$A = b^2 - (b - 2t)^2$$

$$J = (b - t)^3 t$$

$$I_2 = I_3 = \frac{1}{12} [b^4 - (b - 2t)^4]$$

$$SF_2 = SF_3 = 0.44$$

BPROP # 2

$$b = 4.0, t = 0.09$$

$$A = (4)^2 - (4 - 2 \times 0.09)^2 = 1.408$$

$$J = (4 - 0.09)^3 (0.09) = 5.380$$

$$I_2 = I_3 = \frac{1}{12} [(4)^4 - (4 - 2 \times 0.09)^4] \\ = 3.588$$

BPROP # 4

$$b = 2.0, t = 0.125$$

$$A = (2)^2 - (2 - 2 \times 0.125)^2 = 0.938$$

$$J = (2 - 0.125)^3 (0.125) = 0.824$$

$$I_2 = I_3 = \frac{1}{12} [(2)^4 - (2 - 2 \times 0.125)^4] \\ = 0.552$$

PROFORMA

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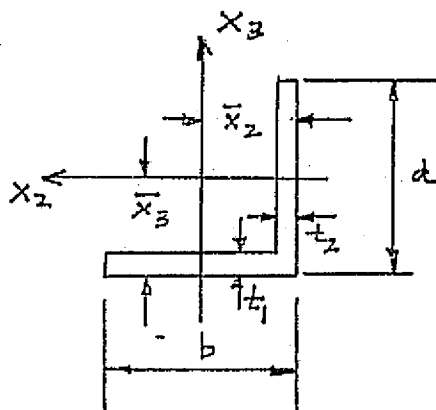
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$$A = d t_2 + (b - t_2) t_1$$

$$\bar{X}_2 = \frac{b^2 t_1 / 2 + (d - t_1) t_2^2 / 2}{A}$$

$$\bar{X}_3 = \frac{d^2 t_2 / 2 + (b - t_2) t_1^2 / 2}{A}$$

$$J = \beta_1 d t_2^3 + \beta_2 (b - t_2) t_1^3$$

$$I_2 = \frac{t_2 d^3}{12} + \frac{(b - t_2) t_1^3}{12} + d t_2 \left(\frac{d}{2} - \bar{X}_3 \right)^2 + (b - t_2) t_1 \left(\bar{X}_3 - \frac{t_1}{2} \right)^2$$

$$I_3 = \frac{t_1 b^3}{12} + \frac{(d - t_1) t_2^3}{12} + b t_1 \left(\frac{b}{2} - \bar{X}_2 \right)^2 + (d - t_1) t_2 \left(\bar{X}_2 - \frac{t_2}{2} \right)^2$$

BPROP # 3

$$b = d = 1.0, t_1 = t_2 = 0.09, \beta_1 = \beta_2 = .33$$

$$\bar{X}_2 = \bar{X}_3 = 0.2829$$

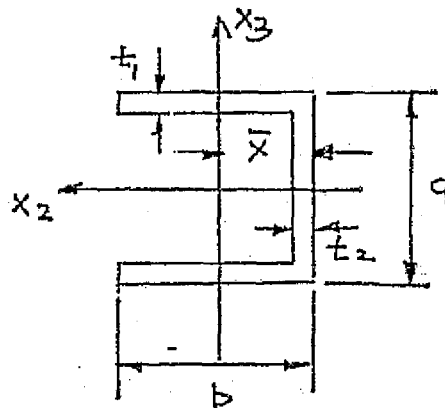
$$I_2 = I_3 = 0.01637$$

$$J = 0.0004589$$

$$SF2 = SF3 = 0.44$$

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$$A = bd - (b - t_2)(d - 2t_1)$$

$$J = \beta_1 d t_2^3 + 2\beta_2 (b - t_2) t_1^3$$

$$I_2 = \frac{1}{12} [bd^3 - (b - t_2)(d - 2t_1)^3]$$

$$I_3 = \frac{2t_1 b^3}{12} + \frac{(d - 2t_1)t_2^3}{12} + 2bt_1 \left(\frac{b}{2} - \bar{x} \right)^2 + t_2 (d - 2t_1) \left(\bar{x} - \frac{t_2}{2} \right)^2$$

BPROP # 5

$$b = d = 2.0, t_1 = t_2 = .125, \beta_1 = \beta_2 = .33$$

$$A = 0.7187$$

$$J = 0.00389$$

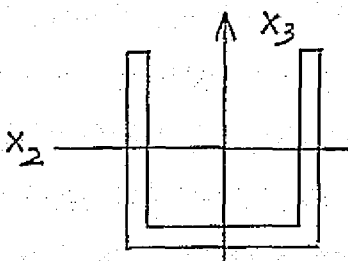
$$I_2 = 0.4959$$

$$I_3 = 0.300$$

$$SF_2 = .33$$

$$SF_3 = .66$$

BPROP # 8



$$b = d = 0.75, t_1 = t_2 = .09, \beta_1 = \beta_2 = .33$$

$$A = 0.1863$$

$$J = .000558$$

$$I_2 = 0.01036$$

$$I_3 = 0.01618$$

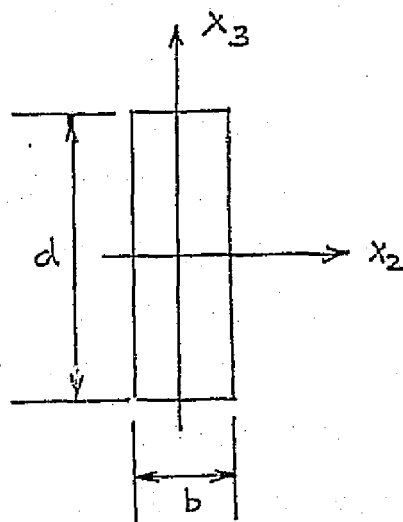
$$SF_2 = .66$$

$$SF_3 = .33$$

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$$A = bd$$

$$J = db^3 \left[\frac{1}{3} - 0.21 \frac{b}{d} \left(1 - \frac{b^4}{12d^4} \right) \right]$$

$$I_2 = bd^3/12$$

$$I_3 = db^3/12$$

BPROP # 6

$$A = 0.4725$$

$$J = 0.00125$$

$$I_2 = 0.2067$$

$$I_3 = 0.0003189$$

$$SF2 = SF3 = 0.44$$

INCOB

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BPROP#7 SIDE SUPPORT STRUT

AREA , $A = 3.3 \text{ IN}^2$

YOUNG'S MODULUS , $\gamma = 15 \times 10^6 \text{ PSI}$

DENSITY = 0.065 lb/cu. IN

POISSON RATIO = 0.25

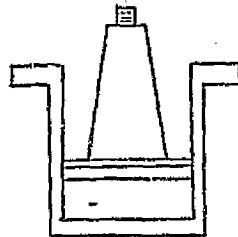
$\alpha = 0.1 \times 10^{-6}$

SIDE SUPPORT STRUT IS AN AXIAL ELEMENT
AS ALL OTHER PROPERTIES ARE UNIMPORTANT.

HENCE, DUMMY VALUES ARE GIVEN.

EQUIVALENT FLEX ARM STIFFNESS

FROM FLEX ARM ANALYSIS:

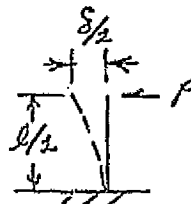
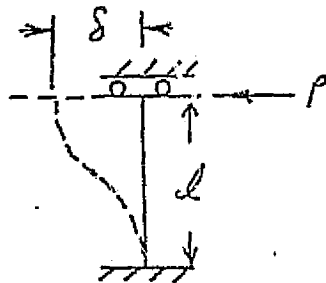


141 LB PRODUCES .020" DEFLECTION.

BLADE THICKNESS:
 $t = .282"$

AVG. WIDTH: $\bar{w} = .75"$

STIFFNESS AS GUIDED CANTILEVER:



$$\delta = .020 \text{ in}$$

$$P = 141 \text{ LB}$$

$$l = 3.375 \text{ in}$$

• FLEX DIRECTION:

$$\frac{\delta}{2} = \frac{P(l/2)^3}{3EI_F} \Rightarrow I_F = \frac{Pl^3}{12\delta}$$

$$\therefore I_F = \frac{(141)(3.375)^3}{12(16 \times 10^6)(.020)} = \underline{\underline{1.41 \times 10^{-3} \text{ in}^4}}$$

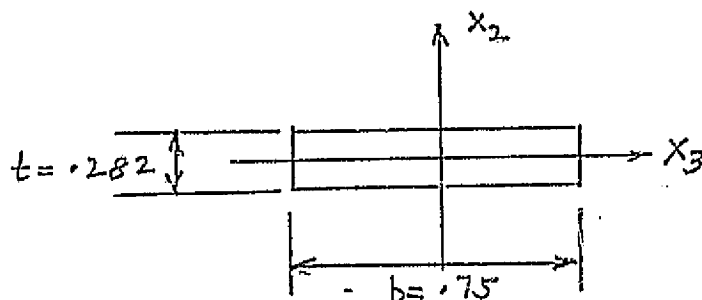
• STIFF DIRECTION:

$$I_S = \frac{t\bar{w}^3}{12} = \frac{(.282)(.75)^3}{12} = \underline{\underline{.010 \text{ in}^4}}$$

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BPROP # 9 FLEX ARM



$$I_2 = I_S = 0.010$$

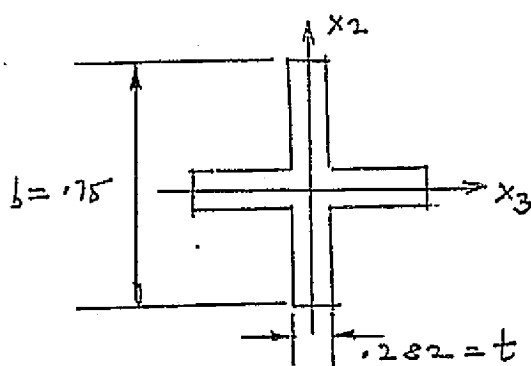
$$I_3 = I_F = 1.41 \times 10^{-3}$$

$$J = bt^3 \left[\frac{1}{3} - 0.21 \frac{t}{b} \left(1 - \frac{t^4}{12bt} \right) \right] = 0.00427$$

$$A = 0.2115$$

$$SF_2 = SF_3 = 0.85$$

BPROP # 10 FIXED ARM



$$\beta = .33$$

$$A = 0.3434$$

$$I_2 = I_3 = I_F + I_S = 0.0114$$

$$J = 2 \left[\beta \times bt^3 + \beta \times (b-t)t^3 \right]$$

$$= 0.0124$$

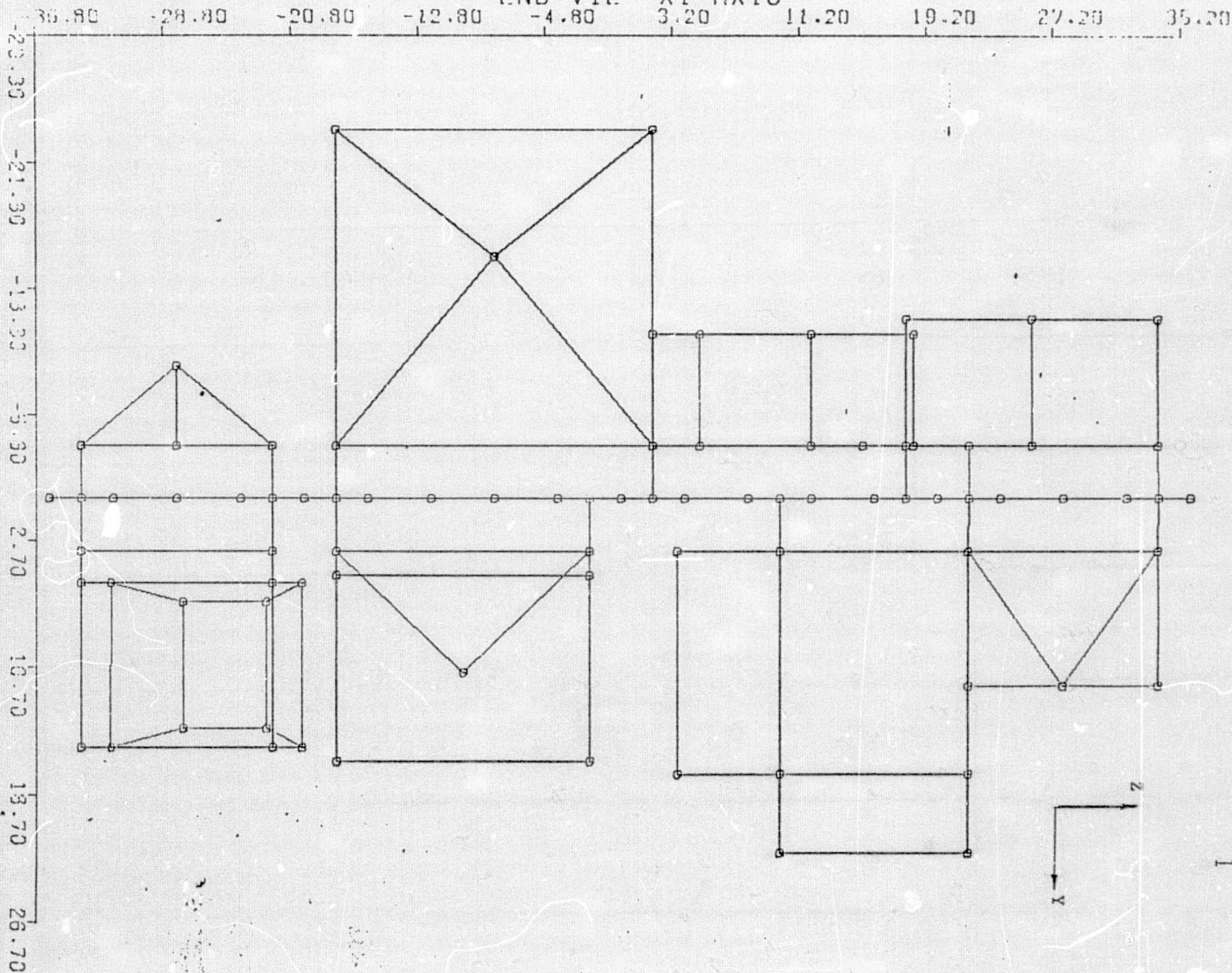
$$SF_2 = SF_3 = 0.615$$

SECTION 6

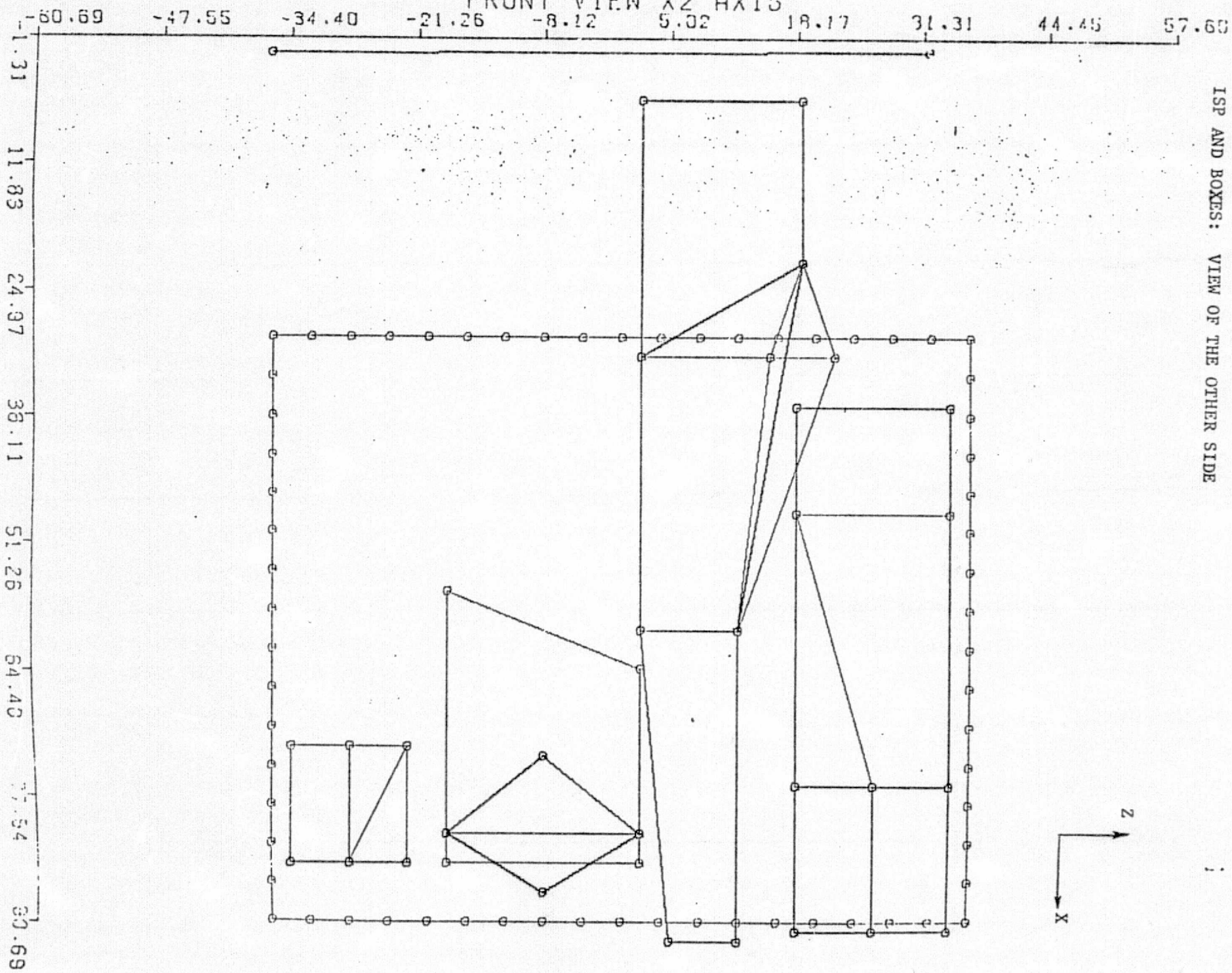
COMPUTER PLOTS OF PANEL AND BOXES

END VIEW XI AXIS

ISP AND BOXES: VIEW ALONG X AXIS

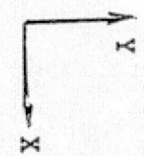


FRONT VIEW X2 AXIS

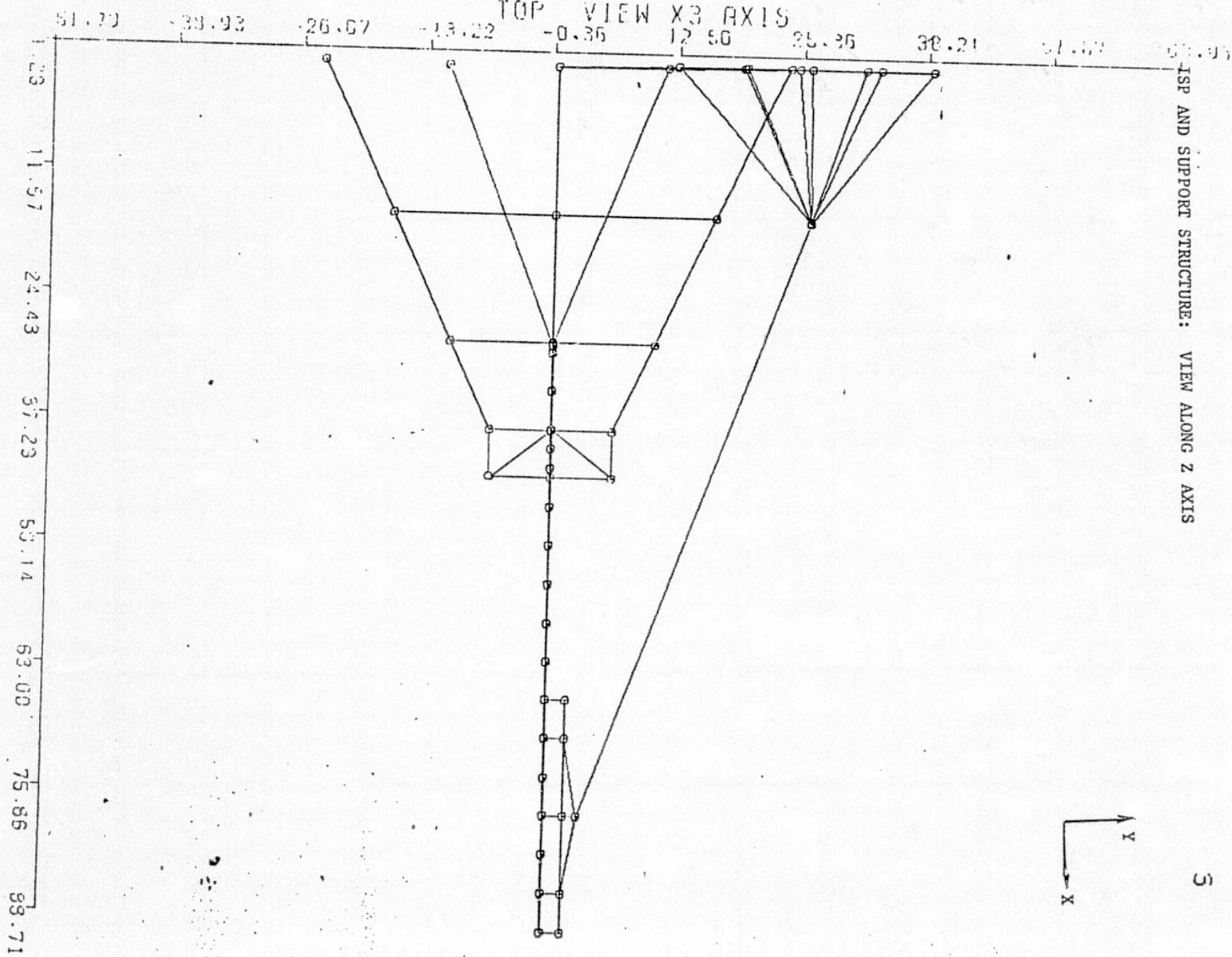


ISP AND BOXES: VIEW OF THE OTHER SIDE

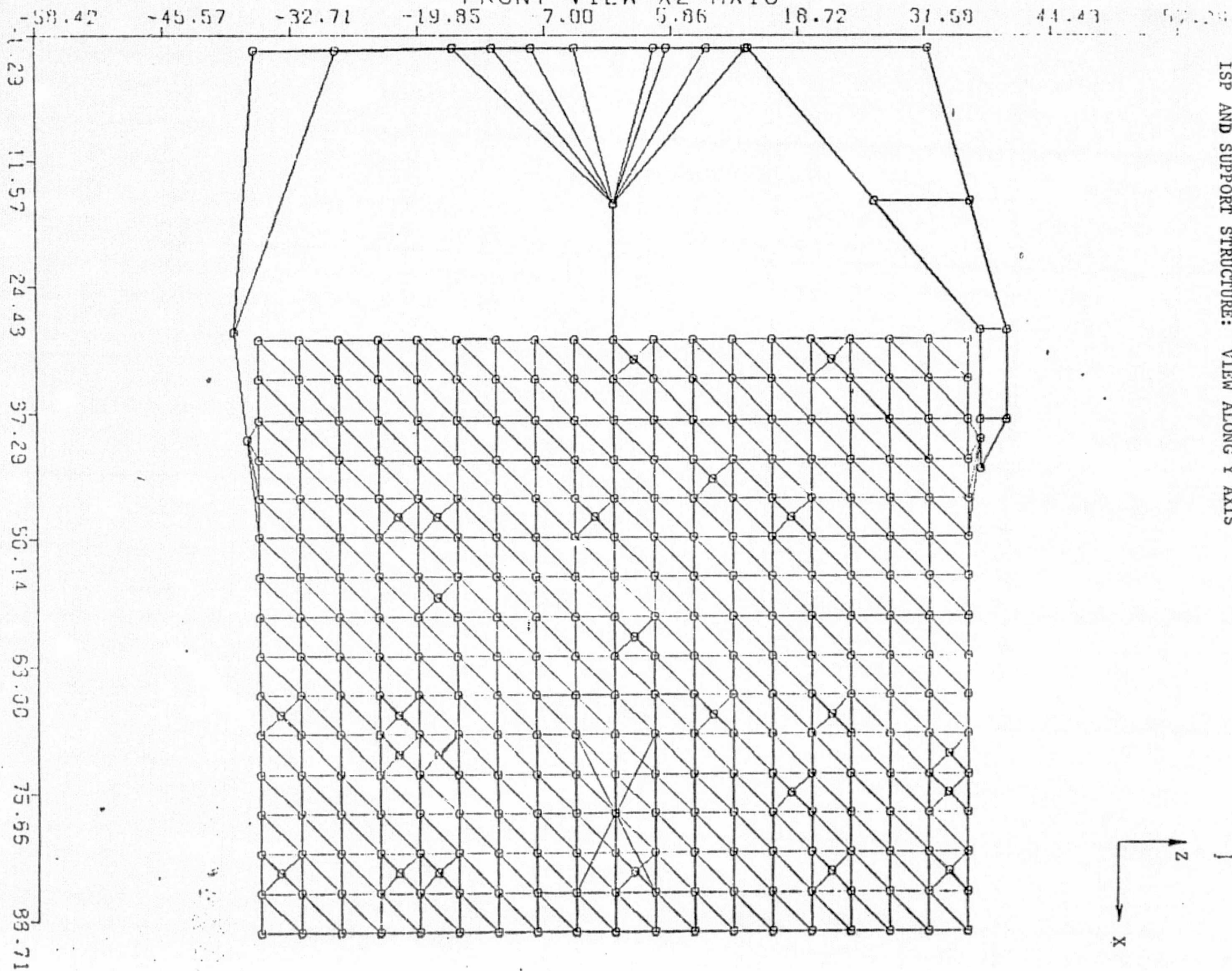
ISP AND SUPPORT STRUCTURE: VIEW ALONG Z AXIS



TOP VIEW X3 AXIS



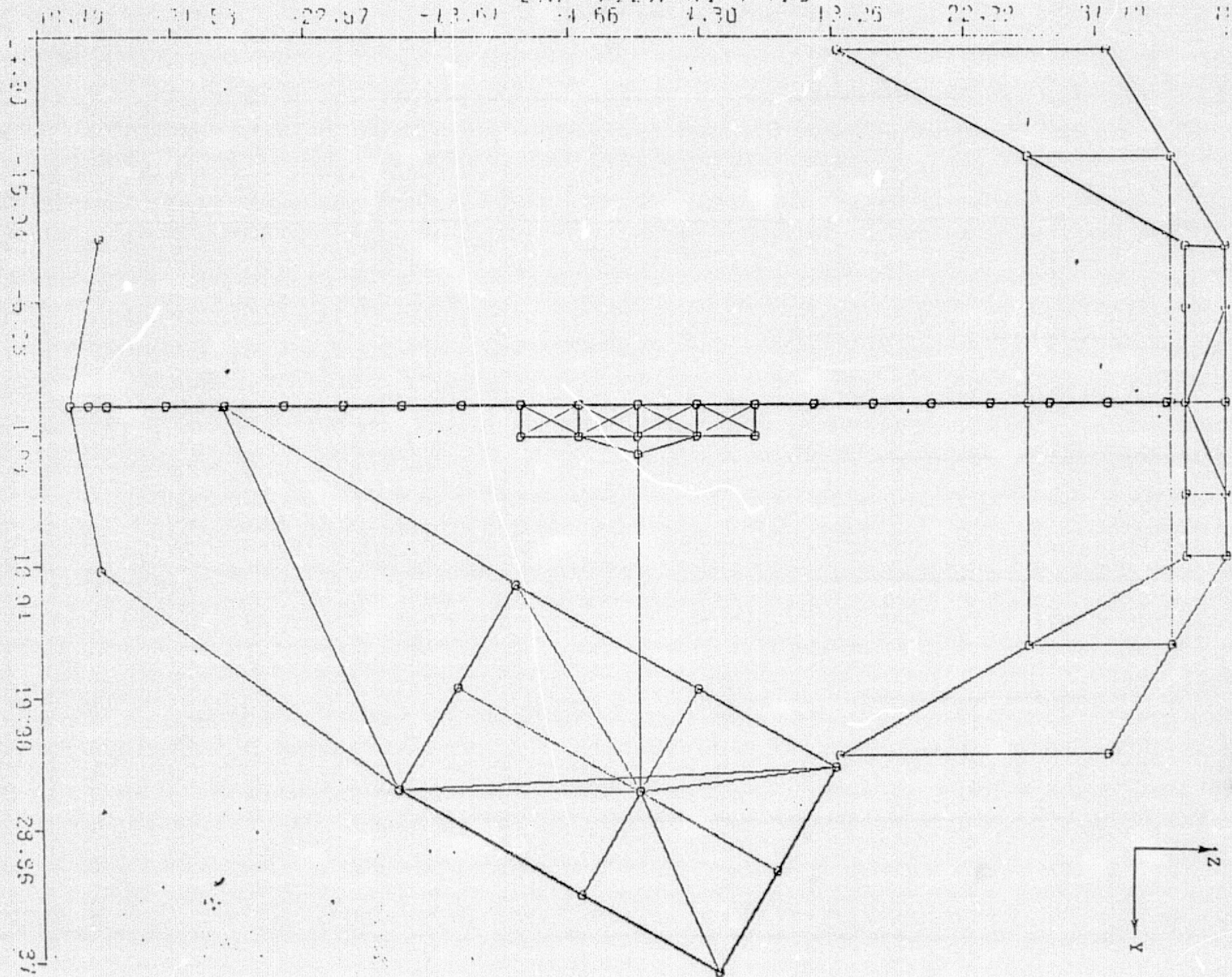
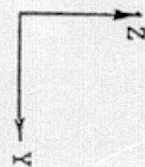
FRONT VIEW X2 AXIS



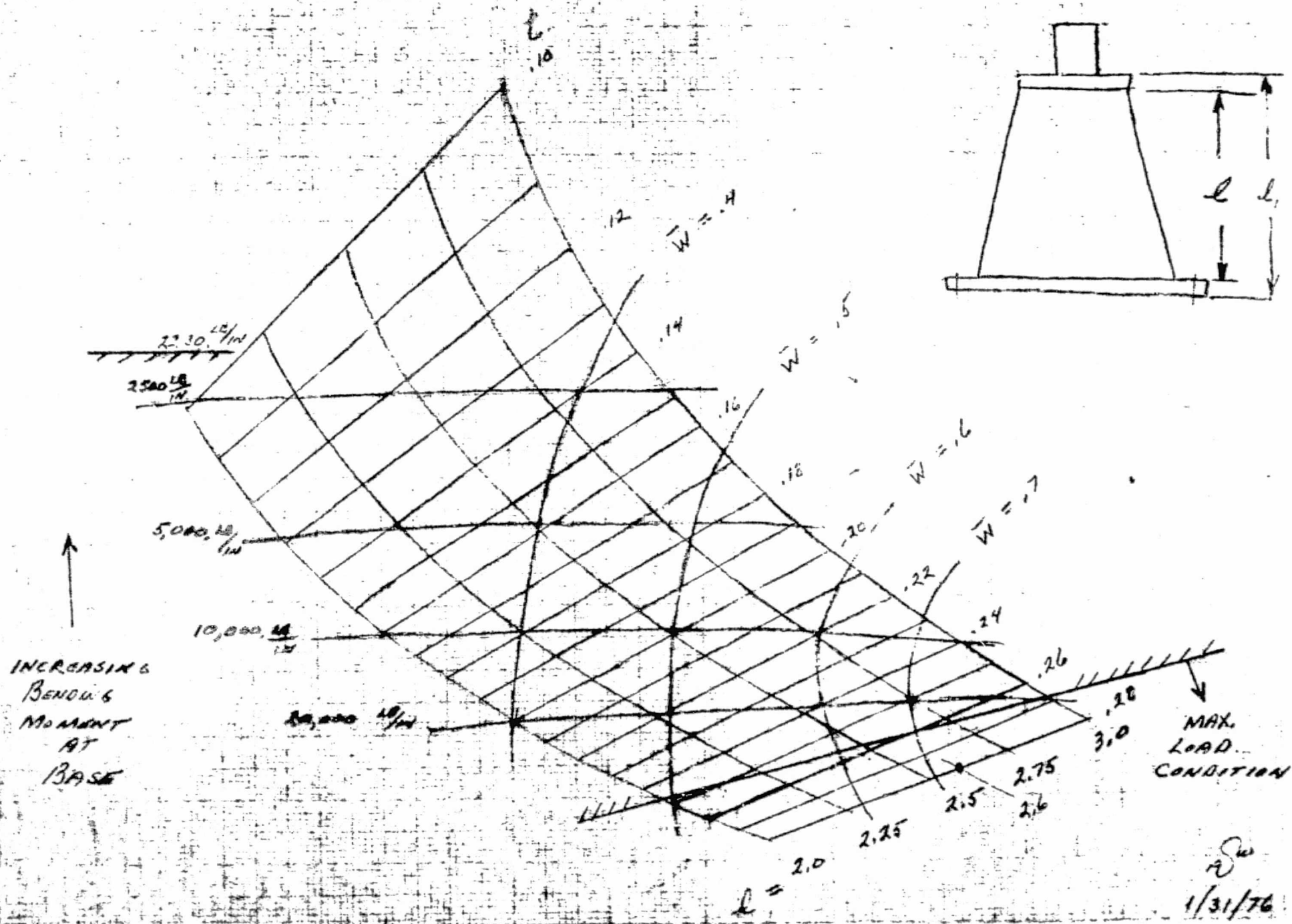
END VIEW XI AXIS

ISP AND STRUCTURE: VIEW ALONG X AXIS

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Flex Beam Motion Chart



APPENDIX E

FLEX MOUNT ASSEMBLY STRESS ANALYSIS



MA 75-12-4

To R. Packer

Location MS 87

Date 7 July 1975

From F. Hayes

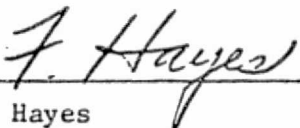
Location MS 105

Telephone 2985

Subject SMM Flex Mount Assembly Stress Analysis

A detailed stress analysis was performed for the Solar Maximum Mission flex mount design for a 2500 lb lateral load condition. A summary of items analyzed is given on page 3. Material yield strengths and load capabilities are shown on page 4.

The detailed analysis of each item is presented in the appendix.


F. Hayes

jah

SOLAR MAX MISSION--FLEX MOUNT ASSEMBLY

Stress Analyses

1. Shear in Mounting Screw.
2. Bearing of Ball Bearing onto Ball Seat.
3. Bearing of Ball Seat onto Mounting Stud.
4. Bending in Flex Mount.
5. Tension in Flex Mount Bolts.
6. Shear in Flex Mount-to-Housing Dowels.
7. Bending in Flex Mount-to-Housing Dowels.
8. Bearing in Flex Mount (Dowel Pins).
9. Bearing in Housing (Dowel Pins).
10. Bending at Base of Housing.
11. Tension in Housing Mounting Screws.
12. Shear in Housing to ISP Dowel Pins.
13. Bearing in Housing (Dowel Pin).
14. Bearing in ISP Face Sheets (Dowel Pin).
15. Bending in ISP Face Sheets.
16. Corner Bracket Screw Loads--Tension.
17. Corner Bracket--Required Friction Loads.

All stresses except screw threads to be below the microyield of the material.

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Subject _____ Page No. _____

STRESS ANALYSIS

No.	TYPE LOADING/LOCATION	MARGIN OF SAFETY	CRITICAL LOAD CONDITION
1	SHEAR/MOUNTING SCREW	+ .74	2500LB X LATERAL
2	BEARING/BALLSEAT	+ .27	..
3	BEARING/MOUNTING SCREW	+ .79	..
4	BENDING/FLEX MOUNT	+ .31	..
5	TENSION/FLEX MOUNT BOLTS	+ .10 ⁽²⁾	..
6	SHEAR/FLEX MOUNT-TO-HOUSING DOWELS	+ .76	..
7	BENDING/FLEX MOUNT-TO-HOUSING DOWELS	+ .13	..
8	BEARING/FLEX MOUNT	- .38 (+ .69) ⁽³⁾	..
9	BEARING/HOUSING	- .88 (AMPLE) ⁽⁴⁾	..
10	BENDING/HOUSING	- .16 (+ .22) ⁽¹⁾	..
11	TENSION/HOUSING MOUNTING SCREWS	AMPLE ⁽²⁾	..
12	SHEAR/HOUSING-TO-ISP DOWEL PINS	- .95 (+ .17) ⁽²⁾	..
13	BEARING/HOUSING	- .92 (AMPLE) ⁽³⁾	..
14	BEARING/IMP FACE SHEETS	- .75 (AMPLE) ⁽³⁾	..
15	BENDING/IMP FACE SHEETS	+ .89	..
16	TENSION/CORNER BRACKET SCREWS	+ 5.4	..
17	SHEAR/CORNER BRACKET SCREWS	- .53 (+ .57) ⁽²⁾	..

(1) 5 MIL INCREASE IN LOWER PORTION OF HOUSING THICKNESS

(2) NOT BASED ON MICHYIELD

(3) INCLUDES FRICTION OF ONE ADJACENT SCREW

(4) INCLUDES FRICTION OF TWO ADJACENT SCREWS

7/3/75

TABLE OF MATERIAL YIELD STRENGTH AND LOAD CAPACITIES

	MATERIAL	SIZE	YIELD(PSI)	MICRO-YIELD (PSI)	TENSILE LOAD CAPACITY (LB)	SHEAR CAPACITY (LB)
	TI-6AL-4V	3/8		70000		
(1)	STAINLESS STEEL A-286 SERIES (BOLTS)	5/16-24	85000	42500	3944	
		3/16-32	85000	42500	1360	
(1)	STAINLESS STEEL 300 SERIES	(#10) 3/16-32	30000	15000	480	-
	STAINLESS STEEL (DOWELS)	5/16 ⁽²⁾	⁽¹⁾	⁽⁴⁾	-	5500
		3/16	60000	30000	-	2000
		1/8	MIN	MIN	-	900
(1)	2024-T6 AL	-	50000	25000	-	-
(3)	6061-T6 AL	-		19000	-	-

(1) per RCA STANDARDS MANUAL

(2) per MIL STANDARD 1516555

(3) per MICROYIELD TABLE 4 FROM BATELLE

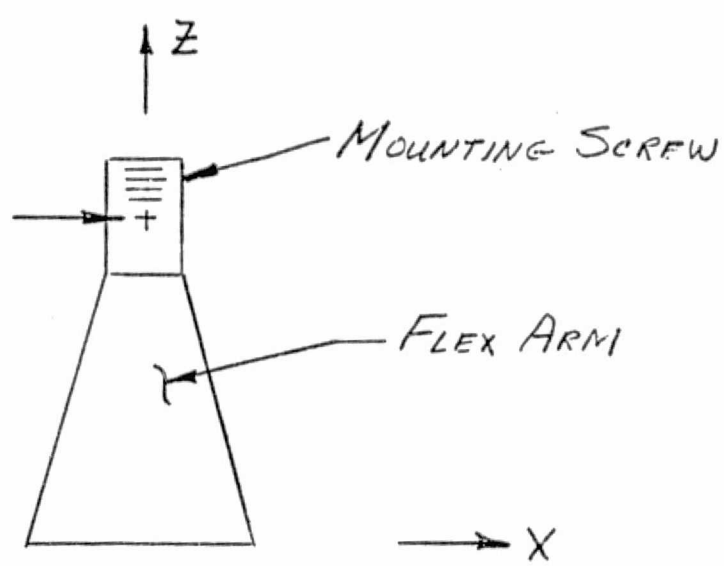
(4) per MATERIALS SELECTOR GUIDE (PRODUCT DESIGN)

Part No. 1-1
Date 7/3/75

7/3/75

1-1
5MM

SHEAR IN MOUNTING STUD



APPLIED LOAD X LATERAL = 2500 LB

SHEAR CAPACITY = LB (3/8 TI-6AL-4V)

$F_{TY} = 70000 \text{ PSI}$, $A_s = .0775 \text{ IN}^2$

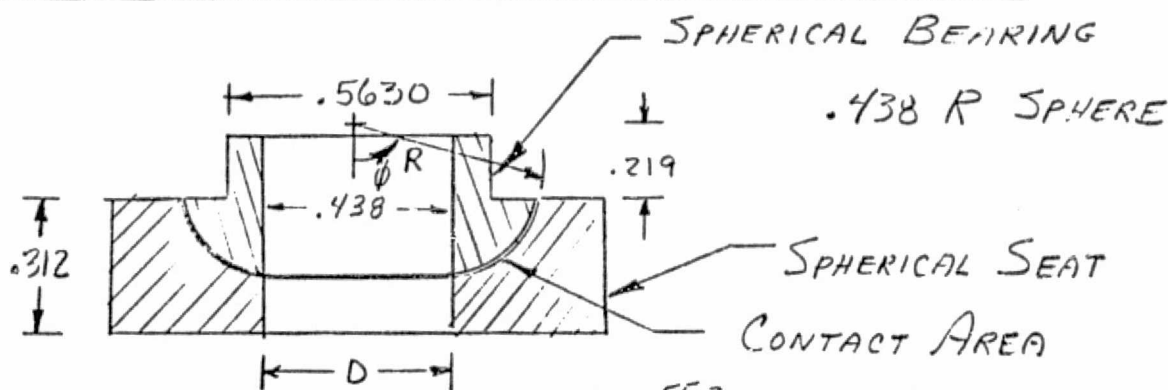
$$S_s = \frac{2500}{.0775} = 32,260 \text{ PSI}$$

$$M.S. = \frac{70000}{(32,260)(1.25)} - 1 = +.74$$

2-1
SAIN

7/3/75

BEARING OF BALL ONTO SPHERICAL SEAT



$$\begin{aligned}
 \text{CONTACT AREA} &= D \left[\int_{30}^{55.2} R \sin \phi d\phi \right] \\
 &= DR \left[-\cos \phi \right]_{30}^{55.2} \\
 &= (.438)(.438) [-\cos 55.2 - (-\cos 30)] \\
 &= .057 \text{ IN}^2
 \end{aligned}$$

$$S_b = \frac{P}{A} = \frac{2500}{.057} = 44,130 \text{ PSI}$$

$$F_{cr} = 70000 \text{ PSI (MICRO YIELD, Ti-6Al-4V)}$$

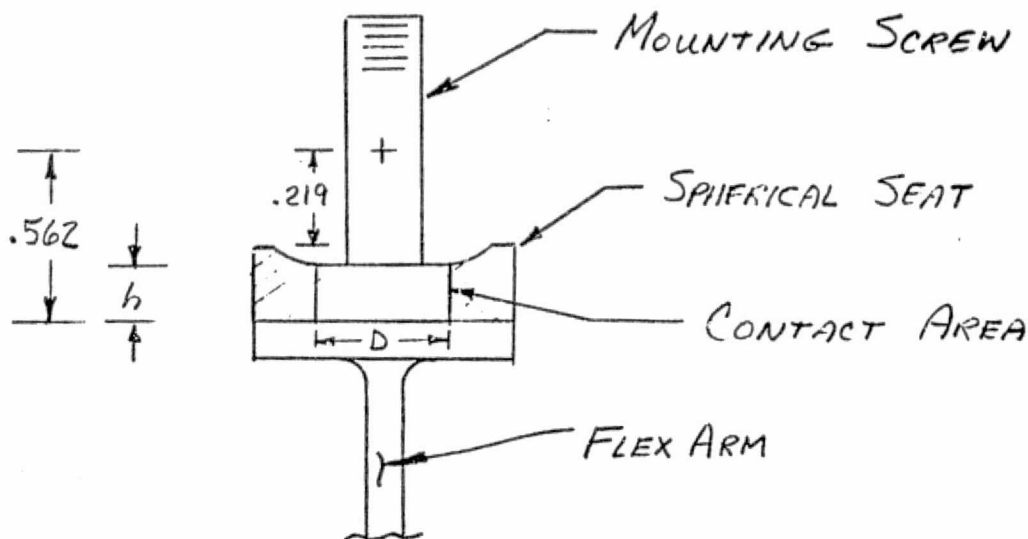
$$M.S. = \frac{70000}{(44130)(1.25)} - 1 = +.27$$

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3-1
5MM

BEARING OF SPHERICAL SEAT ONTO MOUNTING SCREW



$$\text{CONTACT AREA} = D h = (.437)(.183) = .080 \text{ IN}^2$$

FOR 2500 LB LATERAL X LOAD

$$S_b = \frac{P}{A_s} = \frac{2500}{.080} = 31,320 \text{ PSI}$$

$$F_{cr} = 70000 \text{ PSI (MICROYIELD, TI-6AL-4V)}$$

$$M.S. = \frac{70000}{(31,320)(1.25)} - 1 = +.79$$

APPROVED BY
DATE

Prepared for

Report No. 4-1

Approved

Project SMM

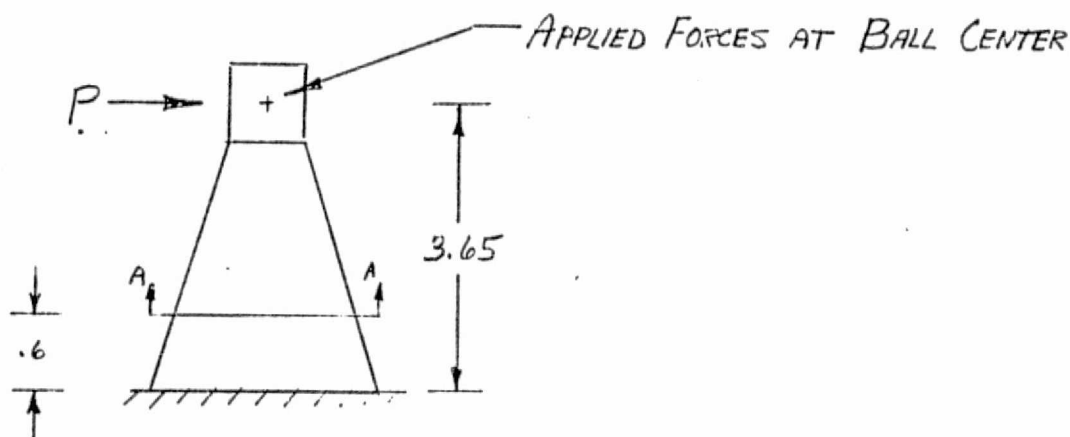
Date 7/2/75

Contract No.

Project No.

Subject

BENDING IN FLEX ARM



$$\sigma_b = \frac{Mc}{I}$$

WITH $\sigma_{\max}$ AT SECTION A-A

where $c = \frac{1.77}{2} = .885$

$$I = \frac{1}{12}(.28)(1.77)^3$$

$$M = 2.5 \text{ IN LB (per FEM)}$$

$$\sigma_b = \frac{(2.5)(.885)}{(.1294)} = 17.10 \text{ PSI (per UNIT LOAD)}$$

APPLIED LOAD X LATERAL: 2500 LB

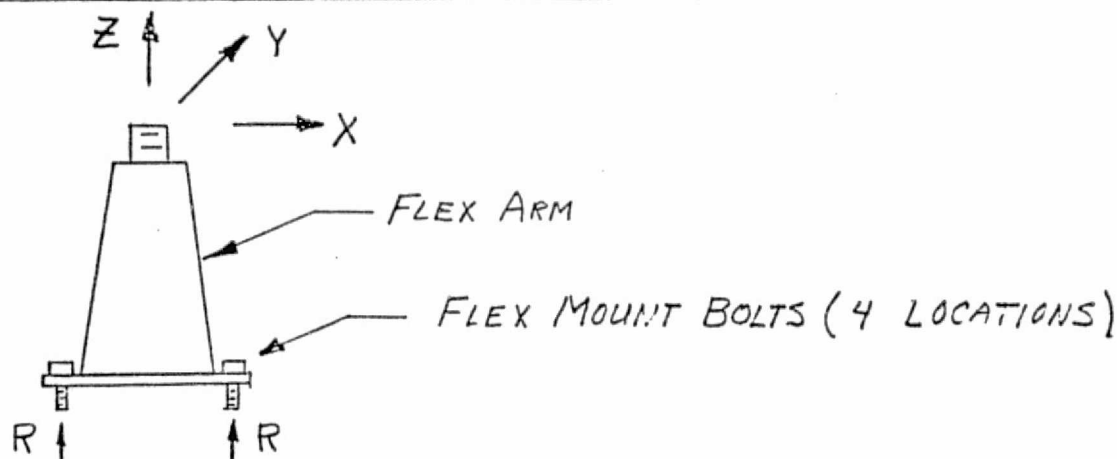
$$\text{MAX STRESS} = 42,750 \text{ PSI}$$

$$F_{TY} = 70000 \text{ PSI (MICRO YIELD)}$$

TI-6AL-4V

$$M.S. = \frac{70000}{(42750)(1.25)} - 1 = +.31$$

TENSION IN FLEX MOUNT BOLTS



R = AXIAL LOAD IN MOUNTING BOLTS

$\left. \begin{matrix} P_x \\ P_y \\ P_z \end{matrix} \right\}$ = APPLIED FORCES AT BALL CENTER

FROM FEM ANALYSIS FOR COMBINED LOADING:

$$\underline{R = 1.24 P_x + 1.28 P_y + .25 P_z}$$

APPLIED LOAD LATERAL $X = 2500$ LB

$$\therefore R = 3100 \text{ LB./BOLT}$$

TENSILE LOAD CAPACITY = 3944 LB ($\frac{5}{16}$ - A-296 SERIES
STAINLESS STEEL BOLT)

$$M.S. = \frac{3944}{(3100)(1.15)} - 1 = +.10 \quad (\text{NOT BASED ON MICROSTRESS})$$

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Drawing

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Approved

Date

Subject

Report No. 6-1

Project

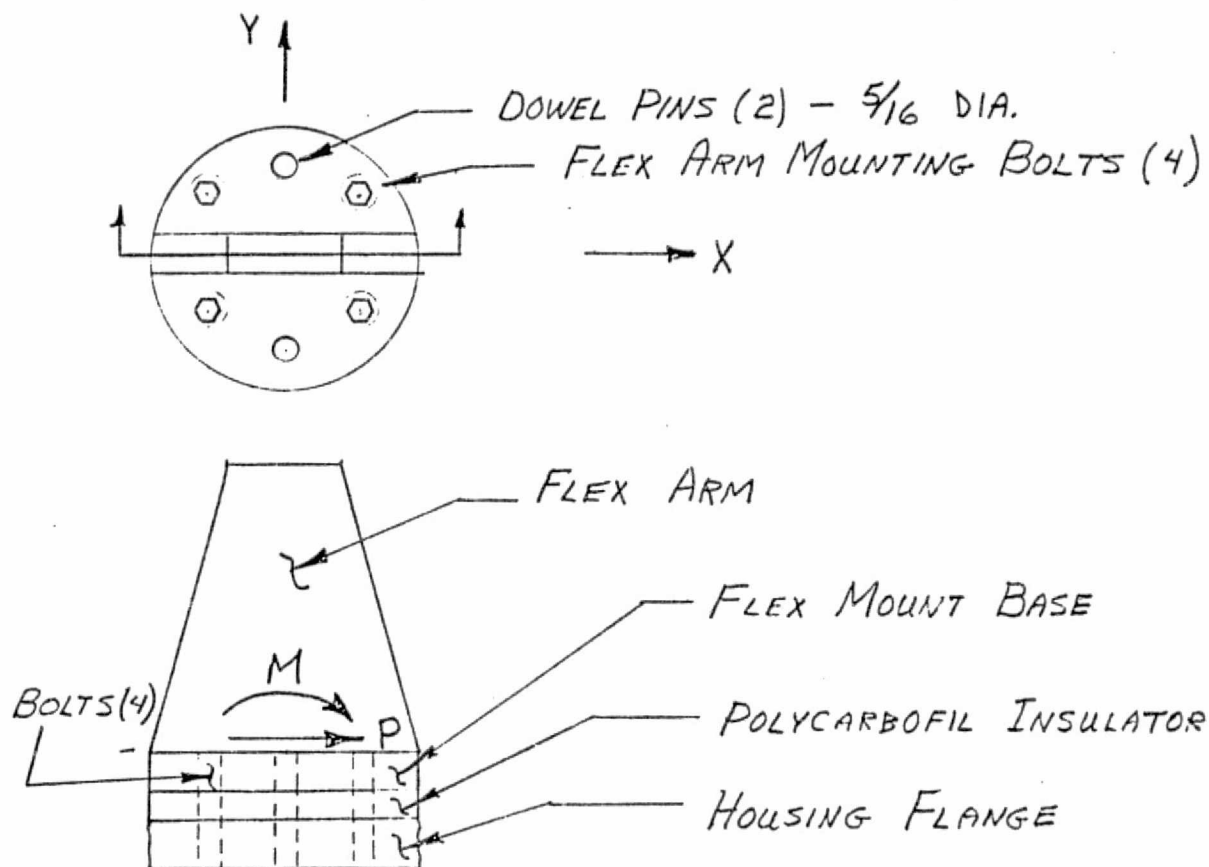
Contract No.

Page No.

SMM

7/3/75

SHEAR IN FLEX MOUNT-TO-HOUSING DOWELS



- ASSUME:
- SHEAR LOAD IS CARRIED BY SHEAR PINS
 - BENDING MOMENT IS CARRIED BY BOLTS (SEE 5-1)

$$P = 1 \text{ LB (per UNIT LOAD)}$$

APPLIED LOAD X LATERAL : 2500 LB

$$\therefore \text{SHEAR FORCE} = 2500 / 2 \text{ PINS} = 1250 \text{ LB (PURE SHEAR)}$$

SHEAR CAPACITY = 5500 LB (5/16 STAINLESS STEEL)

$$M.S. = \frac{5500}{(1250)(1.25)} - 1 = +2.52 \text{ (NOT BASED ON MICROYIELD)}$$

Asst. Dir. of
Tech. Serv.

7/7/75

Report No. 6-2
Project No. SMM
Contract No.

Technical Serv.

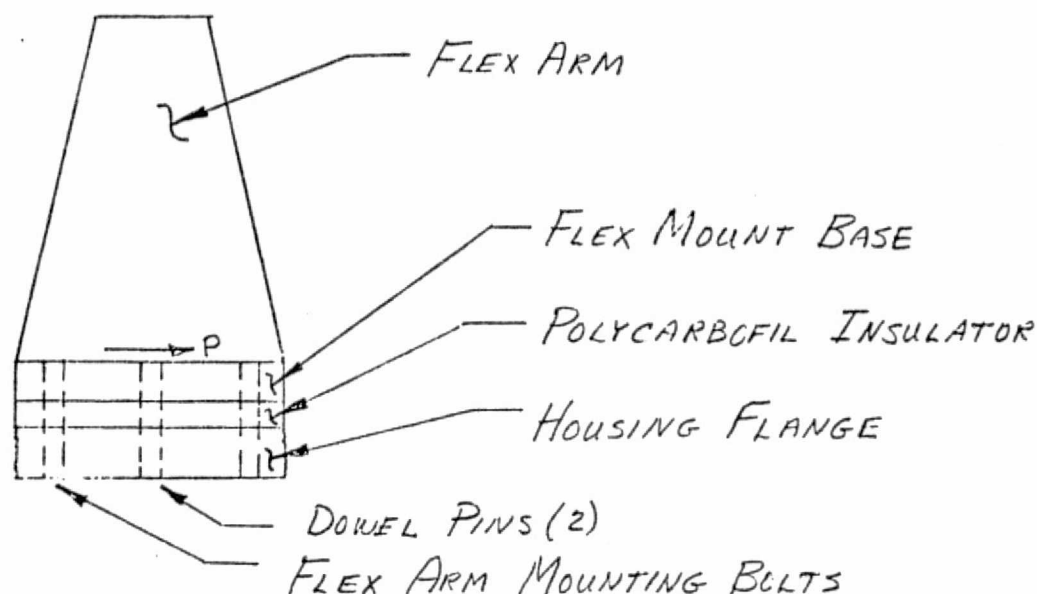
SHEAR IN FLEX MOUNT-TO-HOUSING DOWELS (CONTINUED)

BASED ON MICROYIELD:

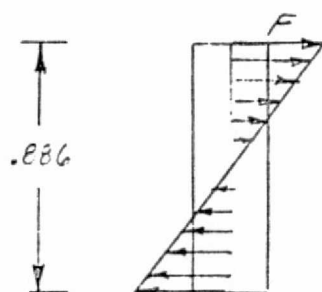
ASSUME MICROYIELD SHEAR CAPACITY IS 50% OF
YIELD SHEAR CAPACITY = $.5(5500) = 2750 \text{ LB}$

$$\therefore M.S. = \frac{2750}{(1250)(1.25)} - 1 = +.76$$

BENDING IN FLEX MOUNT-TO-HOUSING DOWELS



ASSUME: LOCAL BENDING IS DISTRIBUTED AS SHOWN



APPLIED LOAD X LATERAL = 2500 LB

$$F = \frac{3M_p}{2\left(\frac{.886}{2}\right)^2} = \frac{3(.5)(.443)}{2(.443)^2} = 1.69 \text{ LB/IN (PER UNIT LOAD)}$$

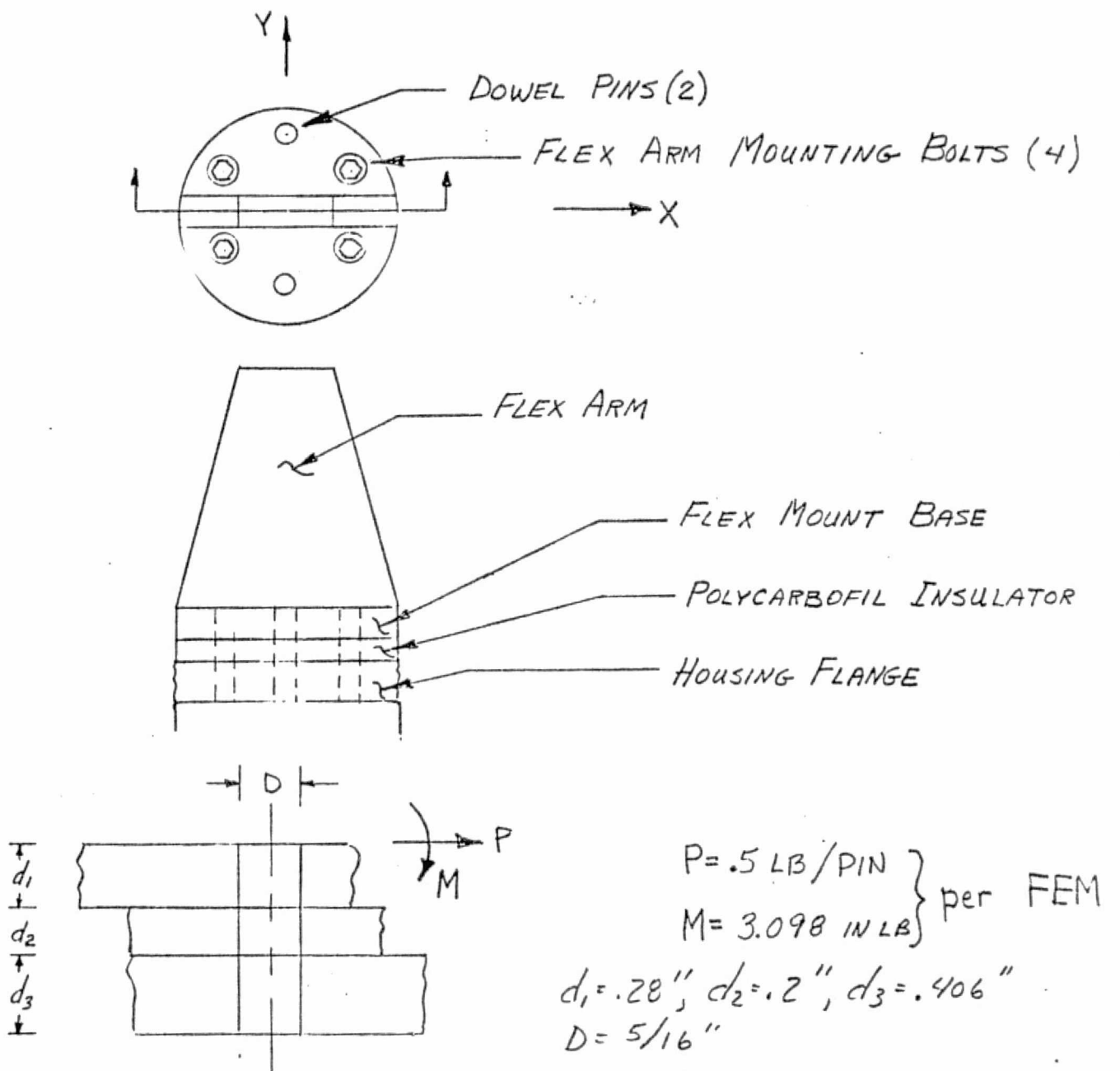
ASSUMING A SINUSOIDAL STRESS DISTRIBUTION AROUND CIRCUMFERENCE (SECT 8-3)

$$\sigma_0 = (2500) \frac{F\pi}{2D} = 2500 \frac{1.69\pi}{2(5/16)} = 21,280 \text{ PSI}$$

$F_{TY} = 30000 \text{ PSI MIN (STAINLESS STEEL, MICROYIELD)}$
 416 SERIES, REF. MTLs. SELECTOR

$$M.S. = \frac{30000}{(21280)(1.25)} - 1 = +.13$$

BEARING IN FLEX MOUNT AT HOUSING INTERFACE:

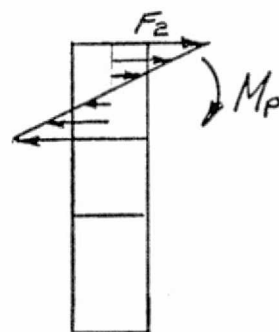
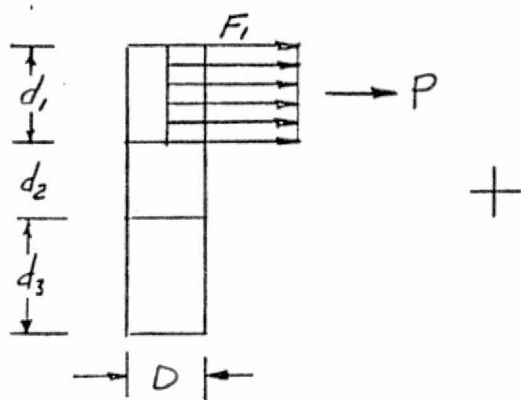


ASSUME: • UNIFORM SHEAR LOAD P AND MOMENT M ACTING AS SHOWN (VALUES OBTAINED FROM FEM COMPUTER MODEL)
 • MOMENT IS CARRIED BY FLEX-MOUNT BOLTS (SECT. 5-1)

BEARING IN FLEX MOUNT AT HOUSING INTERFACE (CONTINUED)

THE FORCE PER UNIT WIDTH ACTING ON THE FLEX MOUNT IS THE SUM OF TWO LOADS:

- BEARING IN FLEX MOUNT DUE TO SHEAR FORCE P
- BEARING IN FLEX MOUNT DUE TO LOCAL BENDING MOMENT CAUSED BY SHEAR FORCE P

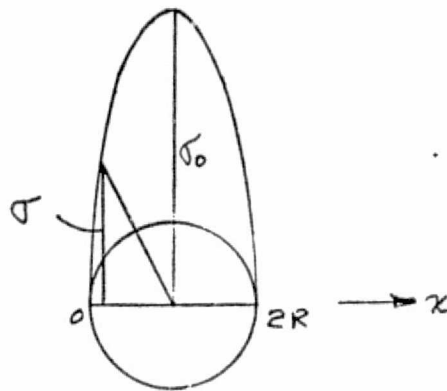


$$\begin{aligned}d_1 &= .28'' \\d_2 &= .2'' \\d_3 &= .406''\end{aligned}$$

$$\begin{aligned}F &= F_1 + F_2 \\&= \frac{P}{d_1} + \frac{3M_p}{2(d_1/2)^2} = \frac{.5}{.28} + \frac{3(.5)(.14)}{2(.14)^2} \\&= 7.14 \text{ lb/in (per UNIT LOAD)}\end{aligned}$$

BEARING IN FLEX MOUNT AT HOUSING INTERFACE (CONTINUED)

ASSUME STRESS VARIES SINUSOIDALLY AROUND CIRCUMFERENCE,



$$\sigma = \sigma_0 \sin \frac{x\pi}{2R}$$

$$\begin{aligned} F &= \int_0^{2R} \sigma dx \\ &= \sigma_0 \int_0^{2R} \sin \frac{x\pi}{2R} dx \\ &= \sigma_0 \frac{2R}{\pi} \left[-\cos \frac{x\pi}{2R} \right]_0^{2R} \\ &= \sigma_0 \frac{2R}{\pi} \left[-(-1) - (-1) \right] = \sigma_0 \frac{4R}{\pi} \end{aligned}$$

$$\therefore \sigma_0 = \frac{F\pi}{4R} = \frac{F\pi}{2D}$$

$$\text{For } D = 5/16", \quad \sigma_0 = \frac{(7.14)(\pi)}{2(5/16)} = 35.90 \text{ PSI (UNIT LOAD)}$$

APPLIED LOAD X LATERAL = 2500 LB

$$\therefore \text{MAX STRESS} = 89,760 \text{ PSI}$$

Astro Electronics
Div. 201

Project No. 8-4

Prepared For

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BEARING IN FLEX MOUNT AT HOUSING INTERFACE (CONTINUED)

$$F_{TY} = 70000 \text{ PSI (MICROYIELD, TI-6AL-4V)}$$

$$M.S. = \frac{70000}{(89760)(1.25)} - 1 = -.38$$

HOWEVER, PART OF THE LOAD IS CARRIED BY THE FRICTION FORCE EXERTED FOR 5/16" TITANIUM BOLTS. TO BE CONSERVATIVE, ASSUME ONLY ONE BOLT IS EFFECTIVE.

$$F_F = (.2)(3944)$$

THUS, THE REMAINING LOAD $P = (.5)(2500) - 394.4 = 461.2 \text{ LB}$

$$\therefore F = \frac{461.2}{.28} + \frac{3(461.2)(.14)}{2(.14)^2} = 6590 \text{ LB/IN}$$

$$\sigma_0 = \frac{F\pi}{2D} = \frac{(6590)\pi}{2(5/16)} = 33120 \text{ PSI}$$

$$M.S. = \frac{70000}{(33120)(1.25)} - 1 = +.69$$

Aero-Electronics
Division

Project No. 9-1

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Project SMM

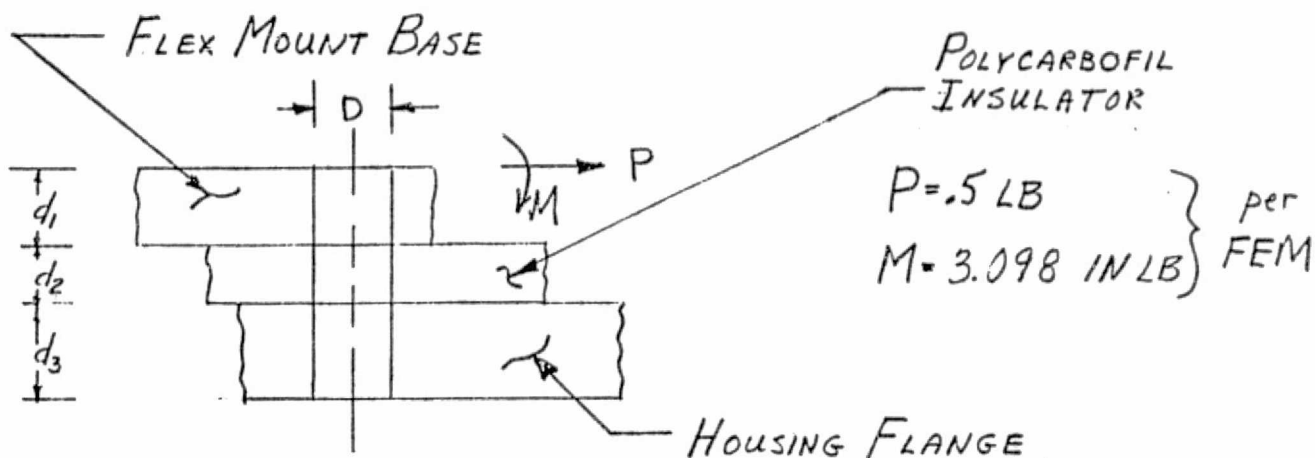
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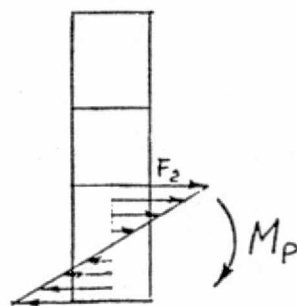
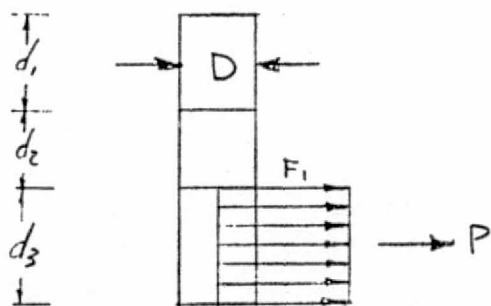
BEARING IN HOUSING AT FLEX MOUNT BASE



- ASSUME:
- UNIFORM SHEAR LOAD P AND MOMENT M ACTING AS SHOWN (VALUES OBTAINED FROM FEM COMPUTER MODEL)
 - MOMENT IS CARRIED BY FLEX MOUNT BOLTS (SECT. 5-1)

THE FORCE/UNIT WIDTH ACTING ON THE HOUSING CONSISTS OF:

- BEARING IN HOUSING FLANGE DUE TO SHEAR FORCE P
- BEARING IN HOUSING FLANGE DUE TO LOCAL BENDING-MOMENT CAUSED BY SHEAR FORCE P



$D = 5/16"$

$d_1 = .23"$

$d_2 = .2"$

$d_3 = .406"$

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BEARING IN HOUSING AT FLEX MOUNT BASE (CONTINUED)

$$\therefore F = \frac{P}{d_3} + \frac{3 M_P}{2(d_3/2)^2} = \frac{.5}{.406} + \frac{3(.5)(.28 + .2 + .406/2)}{2(.203)^2}$$

$$F = 13.66 \text{ LB/IN (per UNIT LOAD)}$$

AGAIN, ASSUMING A SINUSOIDAL STRESS DISTRIBUTION
AROUND THE CIRCUMFERENCE, (SEE SECT 8-3)

$$\sigma_o = \frac{F\pi}{2D} = \frac{(13.66)\pi}{2(5/16)} = 68.67 \text{ PSI (per UNIT LOAD)}$$

APPLIED LOAD X LATERAL = 2500 LB

$$\therefore \text{MAX STRESS} = 172,000 \text{ PSI}$$

$$F_{TY} = 25000 \text{ PSI (MICROYIELD, 2024-T6 Al)}$$

$$M.S. = \frac{25000}{(172,000)(1.25)} - 1 = -.88$$

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BEARING IN HOUSING AT FLEX MOUNT BASE (CONTINUED)

AGAIN, CONSIDER FRICTION FORCE EXERTED BY ONE
5/16 TITANIUM BOLT, $F_F = (.2)(3944) = 788.8 \text{ LB}$

THUS, $P = 461.2 \text{ LB}$

$\therefore F = 12,600 \text{ LB} \Rightarrow \sigma_o = 63,350 \text{ PSI}$

$$\therefore M.S. = \frac{25000}{(63350)(1.25)} - 1 = -.68$$

BY CONSIDERING FRICTION FORCE EXERTED BY TWO
5/16 TITANIUM BOLTS, $F_F = 1577.6 \text{ LB}$ AND
THEREFORE BEARING IN HOUSING SHOULD NOT
OCCUR SINCE THE SHEAR LOAD OF 1250 LB/PIN
IS LESS THAN THE FRICTION FORCE.

$M.S. = \text{AMPLE}$

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App: _____

Date: _____

Subject: _____

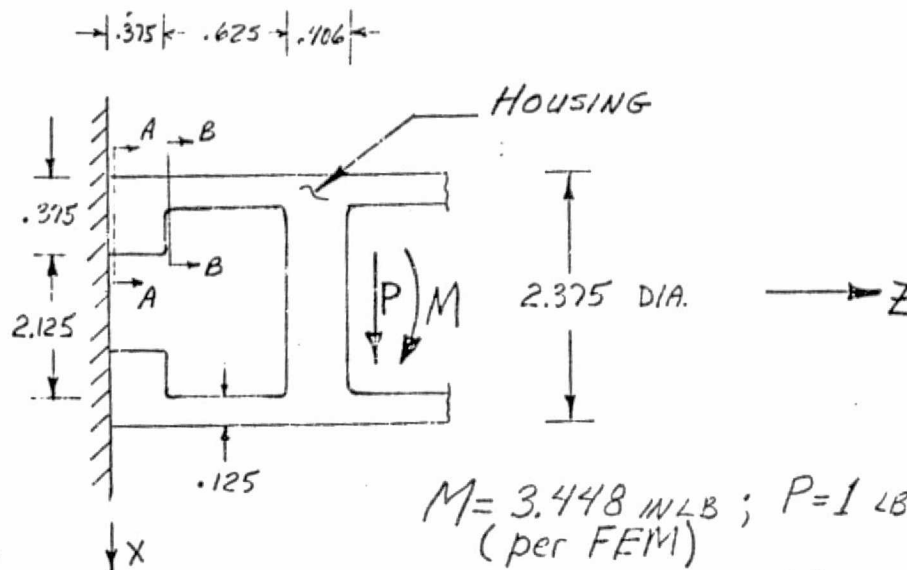
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Revision No.: _____

BENDING AT BASE OF HOUSING



$$M = 3.448 \text{ IN LB ; } P = 1 \text{ LB (per UNIT LOAD)} \\ \text{(per FEM)}$$

ASSUME: • BASE IS CONSTRAINED FROM MOTION

• ALL THE LOAD IS CARRIED BY BENDING IN HOUSING

$$\text{AT SECTION A-A: } M = 3.448 + 1(1.4) = 4.848 \text{ IN LB}$$

$$M = 4.848 \text{ IN LB (per UNIT LOAD)}$$

$$S_b = \frac{Mc}{I}$$

$$\text{where } c = 1.1875 \text{ IN.}$$

$$I = \frac{\pi}{4} [(1.1875)^4 - (.8125)^4]$$

$$= 1.2195 \text{ IN}^4$$

$$\therefore S_b = - \frac{(4.848)(1.1875)}{(1.2195)} = 4.721 \text{ PSI (per UNIT LOAD)}$$

BENDING AT BASE OF HOUSING (CONTINUED)

APPLIED LOAD X LATERAL = 2500 LB

$$\therefore S_b = 11,800 \text{ PSI} \quad F_{TY} = 25000 \text{ PSI (MICROYIELD)}$$

2024-T6 AL

$$M.S. = \frac{25000}{(11800)(1.25)} - 1 = +.67$$

AT SECTION B-B: $M = 3.448 + 1(1.025)$

AT SECTION B-B: $M = 4.473 \text{ IN LB (per UNIT LOAD)}$

$$S_b = \frac{Mc}{I} \quad \text{where} \quad c = 1.1875$$

$$I = \frac{\pi}{4} \left[(1.1875)^4 - (1.0625)^4 \right]$$

$$I = 0.5609 \text{ IN}^4$$

$$\therefore S_b = \frac{(4.473)(1.1875)}{.5609} = 9.4706 \text{ PSI (per UNIT LOAD)}$$

APPLIED LOAD X LATERAL = 2500 LB $\Rightarrow S_b = 23,680 \text{ PSI}$

$F_{TY} = 25000 \text{ PSI (MICROYIELD, 2024-T6 AL)}$

$$M.S. = \frac{25000}{(23680)(1.25)} - 1 = -.16$$

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BENDING IN HOUSING (CONTINUED)

BY INCREASING THICKNESS OF LOWER PORTION OF HOUSING 5 MILS, THERE IS A POSITIVE MARGIN OF SAFETY.

$$I = \frac{\pi}{4} [(1.1875)^4 - (.98875)^4] = .8112$$

$$\therefore S_b = \frac{(4.473)(1.1875)}{.8112}$$

$$M.S. = \frac{25000}{(20465)(1.25)} - 1 = +.22$$

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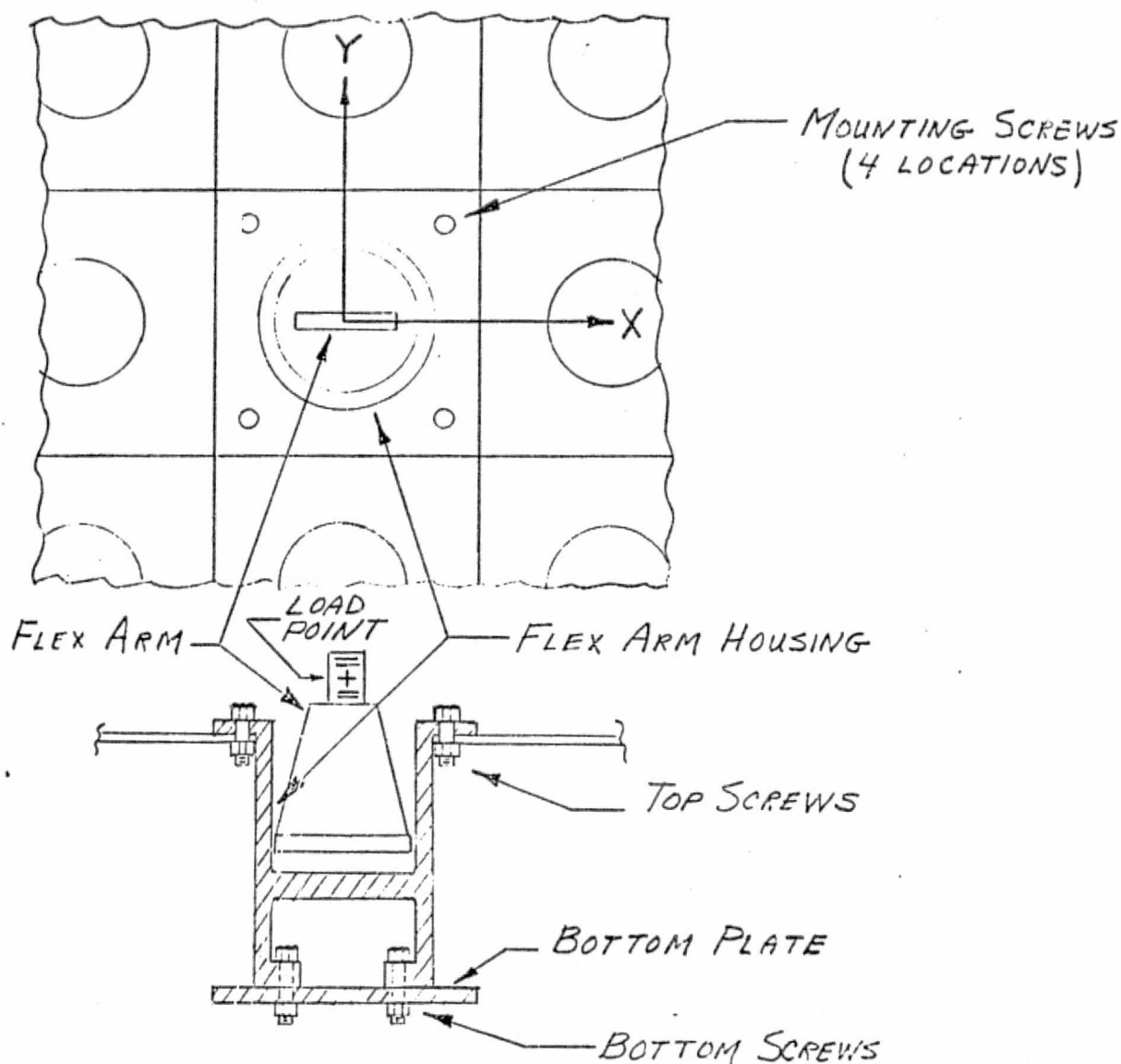
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TENSION IN HOUSING MOUNTING SCREWS



- FIND TENSILE FORCE IN TOP SCREWS
 - FIND TENSILE FORCE IN BOTTOM SCREWS
(NOT INCLUDED SINCE FORCE IS GREATER IN TOP SCREWS)
- } STATIC LOADING
X, Y, Z

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TENSION IN HOUSING MOUNTING SCREWS (CONTINUED)

TOP SCREWS (AXIAL FORCES per UNIT APPLIED LOAD)

+ TENSION - COMPRESSION

LOAD CONDITION: LATERAL X

TENSILE FORCE = .0787412 LB.

LOAD CONDITION: LATERAL Y

TENSILE FORCE = .0595831 LB.

LOAD CONDITION: AXIAL Z

TENSILE FORCE = .1674624 LB.

APPLIED LOAD X = 2500 LB	}	TOTAL FORCE = 197 LB = .0787(2500) = 197.
APPLIED LOAD Y = - LB		
APPLIED LOAD Z = - LB		

LOAD CAPACITY = 3944 LB ($\frac{5}{16}$ -A-286 SERIES
STAINLESS STEEL BOLTS)

M.S. = $\frac{3944}{(197)(1.25)} - 1 = +15$. (AMPLE)
(NOT BASED ON MICKOYIELD)

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Author	
Checker	
Engineer	

TENSION IN HOUSING MOUNTING SCREWS (CONTINUED)

BASED ON MICROYIELD

USE 50% OF TENSILE LOAD CAPACITY FOR YIELD

$$\text{OR } (.50)(3944) = 1972 \text{ LB}$$

$$M.S. = \frac{1972}{(197)(1.25)} - 1 = + 7.0$$

SHEAR IN HOUSING TO ISP DOWEL PINS

TOP SCREWS (SHEAR FORCE per UNIT APPLIED LOAD)

LOAD CONDITION: LATERAL X

$$\begin{aligned}
 \text{Shear Force } X &= .2314627 \\
 \text{Shear Force } Y &= .0830144
 \end{aligned}
 \}$$

RESULTANT SHEAR

$$F_s = .2458991$$

LOAD CONDITION: LATERAL Y

$$\begin{aligned}
 \text{Shear Force } X &= .0688559 \\
 \text{Shear Force } Y &= .1920657
 \end{aligned}
 \}$$

$$F_s = .2040352$$

LOAD CONDITION: AXIAL Z

$$\begin{aligned}
 \text{Shear Force } X &= .0425976 \\
 \text{Shear Force } Y &= .0425976
 \end{aligned}
 \}$$

$$F_s = .0602421$$

APPLIED LOAD $X = 2500 \text{ LB}$

$$\therefore F_s = (.2459)(2500) = 615 \text{ LB}$$

FOR $\frac{1}{8}$ " DOWELS (CORROSION RESISTING STEEL, STAINLESS)

SHEAR STRENGTH IS 900 LB.

$$M.S. = \frac{900}{(615)(1.25)} - 1 = +.17 \quad (\text{NOT BASED ON MICROYIELD})$$

NOTE: FOR CORNERS WITH NO DOWEL PINS, LOAD IS CARRIED BY FRICTION FORCE OF $\frac{5}{16}$ TI BOLTS (788.8 LB)

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SHEAR IN HOUSING TO ISP DOWEL PINS (CONTINUED)

BASED ON MICROYIELD

USE 50% OF SHEAR CAPACITY (BASED ON YIELD)

$$\text{OR } (.50)(900) = 450 \text{ LB}$$

$$\therefore M.S. = \frac{450}{(615)(1.25)} - 1 = -.95$$

Audio Engineering
Division

Project No. 13-1

Prepared for

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Agency

Project SMM

Date

7/1/75

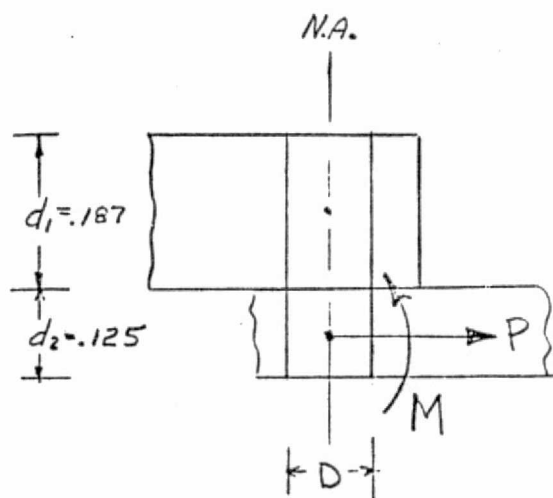
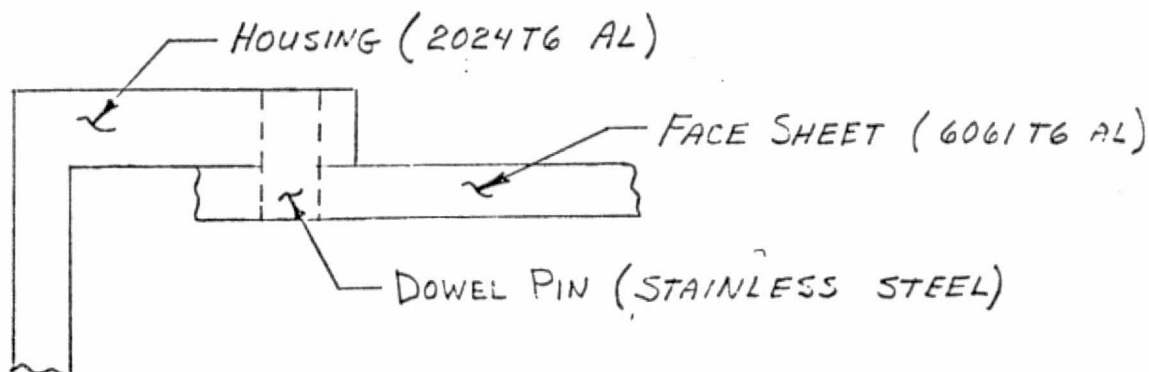
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BEARING IN HOUSING AT ISP INTERFACE

TOP PINS



P = SHEAR FORCE IN DOWEL PIN

M = BENDING MOMENT

D = DIAMETER OF DOWEL PIN

ASSUME: • UNIFORM SHEAR PLUS BENDING MOMENT ACTING
IN FACE SHEET (VALUES OBTAINED FROM FEM COMPUTER RUN)
• MOMENT IS CARRIED BY TENSION & COMPRESSION IN
HOUSING MOUNTING SCREWS (SECT 11-1)

$$P = .2459 \text{ LB/PIN}$$

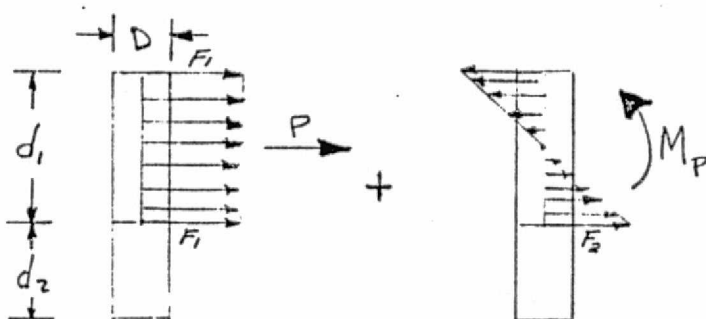
$$M = .0471 \text{ IN LB/}$$

per FEM

BEARING IN HOUSING AT IMP INTERFACE (CONTINUED)

THUS, THE TOTAL FORCE/UNIT WIDTH ACTING ON THE HOUSING CONSISTS OF :

- BEARING IN HOUSING FLANGE DUE TO SHEAR FORCE P
- BEARING IN HOUSING FLANGE DUE TO EQUIVALENT BENDING MOMENT CAUSED BY SHEAR FORCE P



$$\begin{aligned}
 \therefore F &= \frac{P}{d_1} + \frac{3 M_P}{2 (d_1/2)^2} \\
 &= \frac{.2459}{.187} + \frac{3 (.2459) (\frac{.187 + .125}{2})}{2 (.187/2)^2} \\
 &= 7.90 \quad \text{lb/in} \quad (\text{per UNIT LOAD})
 \end{aligned}$$

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BEARING IN HOUSING AT IMP INTERFACE (CONTINUED)

AGAIN, ASSUMING A SINUSOIDAL STRESS DISTRIBUTION
AROUND THE CIRCUMFERENCE,

$$\sigma_o = \frac{F\pi}{2D} = \frac{(7.90)\pi}{2(1/8)} = 99.24 \text{ PSI (per UNIT LOAD)}$$

APPLIED LOAD X LATERAL = 2500 LB

$$\therefore \text{MAX STRESS} = 248,000 \text{ PSI}$$

$$F_{TY} = 25000 \text{ PSI (MICROYIELD, 2024-T6 AL)}$$

$$M.S. = \frac{25000}{(248000)(1.75)} - 1 = -.92$$

CONSIDER FRICTION FORCE EXERTED BY $5/16$ TI BOLTS.
ASSUMING ONE BOLT EFFECTIVE, THE TOTAL SHEAR
LOAD OF 615 LB (SECT 12-1) IS CARRIED BY THE
FRICTION FORCE = $.2(3944) = 788.8 \text{ LB}$.

Aero Electronics
Division
Project No. 14-1

Prepared for

Report No. 14-1

Approved

Project SMM

Date

7/2/75

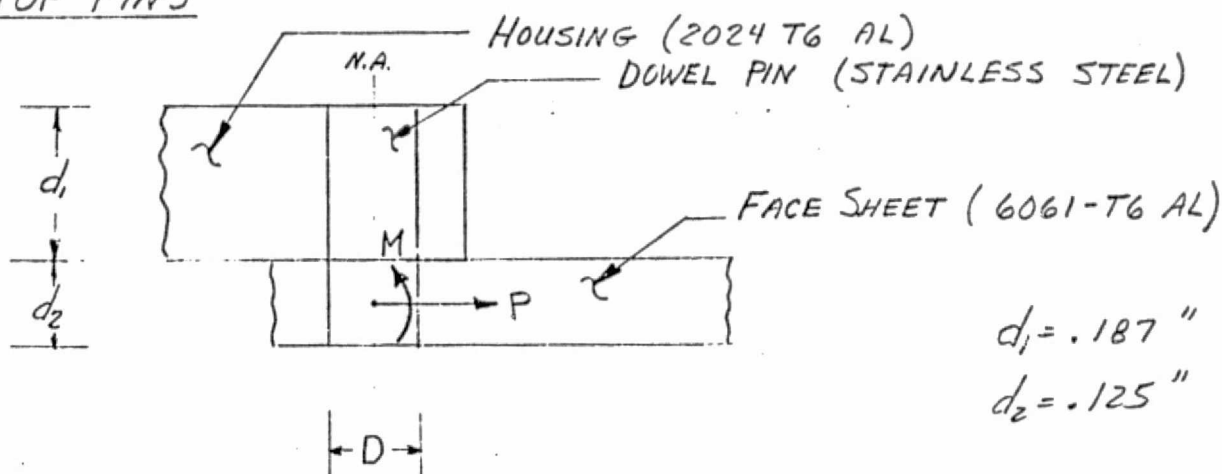
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BEARING IN ISP FACE SHEETS

TOP PINS



ASSUME: • UNIFORM SHEAR PLUS BENDING MOMENT ACTING IN FACE SHEET (VALUES OBTAINED FROM FEM COMPUTER RUN)

• MOMENT IS CARRIED BY TENSION AND COMPRESSION IN HOUSING MOUNTING SCREWS

THE FORCE / UNIT WIDTH ACTING ON THE FACE SHEETS

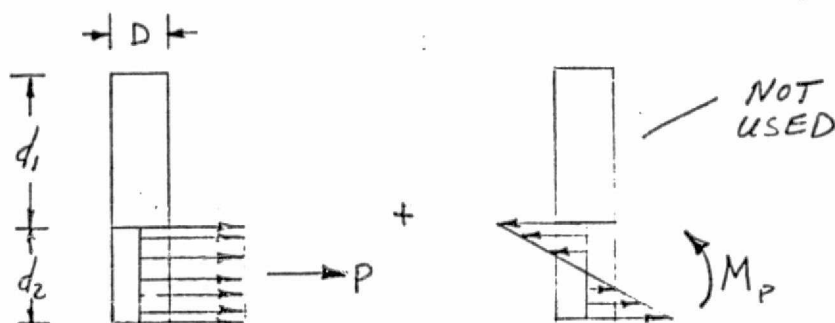
CONSISTS OF:

- BEARING IN FACE SHEET DUE TO SHEAR FORCE P

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BEARING IN IMP FACE SHEETS (CONTINUED)



$$F = \frac{P}{d_2} = \frac{.2459}{.125}$$

$$\therefore F = 1.97 \text{ LB/IN (per UNIT LOAD)}$$

AGAIN, ASSUMING A SINUSOIDAL STRESS DISTRIBUTION
AROUND THE CIRCUMFERENCE,

$$J_0 = \frac{F\pi}{2D} = \frac{(1.97)\pi}{2(1/8)} = 24.72 \text{ PSI (per UNIT LOAD)}$$

$$\text{APPLIED LOAD X LATERAL} = 2500 \text{ LB}$$

$$\therefore \text{MAX. STRESS} = 61,800 \text{ PSI}$$

$$F_{TY} = 19000 \text{ PSI (MICRO YIELD, 6061-T6 AL)}$$

$$M.S. = \frac{19000}{(61800)(1.25)} - 1 = -.75$$

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Location

BEARING IN IMP FACE SHEETS (CONTINUED)

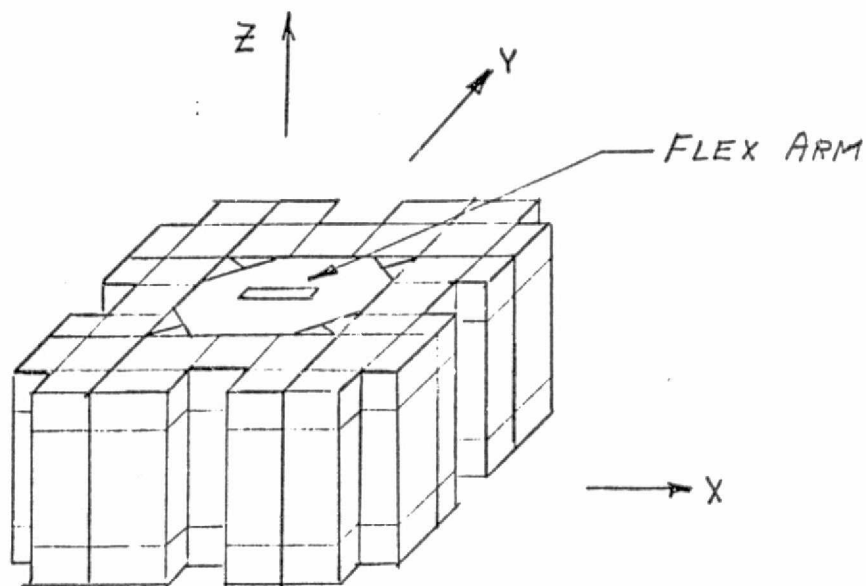
CONSIDER FRICTION FORCE OF 788.8 LB EXERTED
BY 5/16 Ti BOLTS. ASSUMING ONE BOLT EFFECTIVE,
THE TOTAL SHEAR LOAD OF 615 LB IS CARRIED
BY THIS FRICTION FORCE.

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SMM

6/30/75

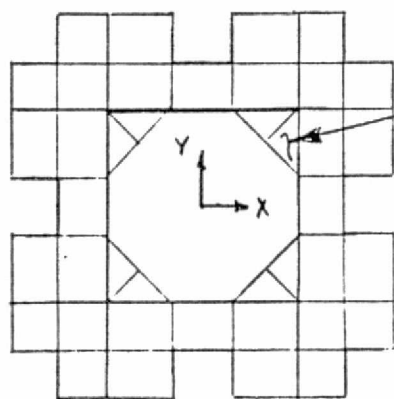
BENDING IN ISP FACE SHEETS

(per UNIT LOAD APPLIED AT FLEX MOUNT)



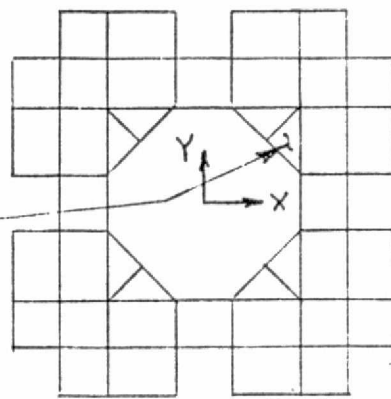
LOAD CONDITION: LATERAL X

TOP FACE SHEET



$\sigma_{MAX} = 3.224 \text{ PSI}$

BOTTOM FACE SHEET



$\sigma_{MAX} = 0.527 \text{ PSI}$

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BENDING IN IMP FACE SHEETS (CONTINUED)

APPLIED LOAD LATERAL $X = 2500$ LB

$\therefore$ MAX. STRESS = 8060 PSI

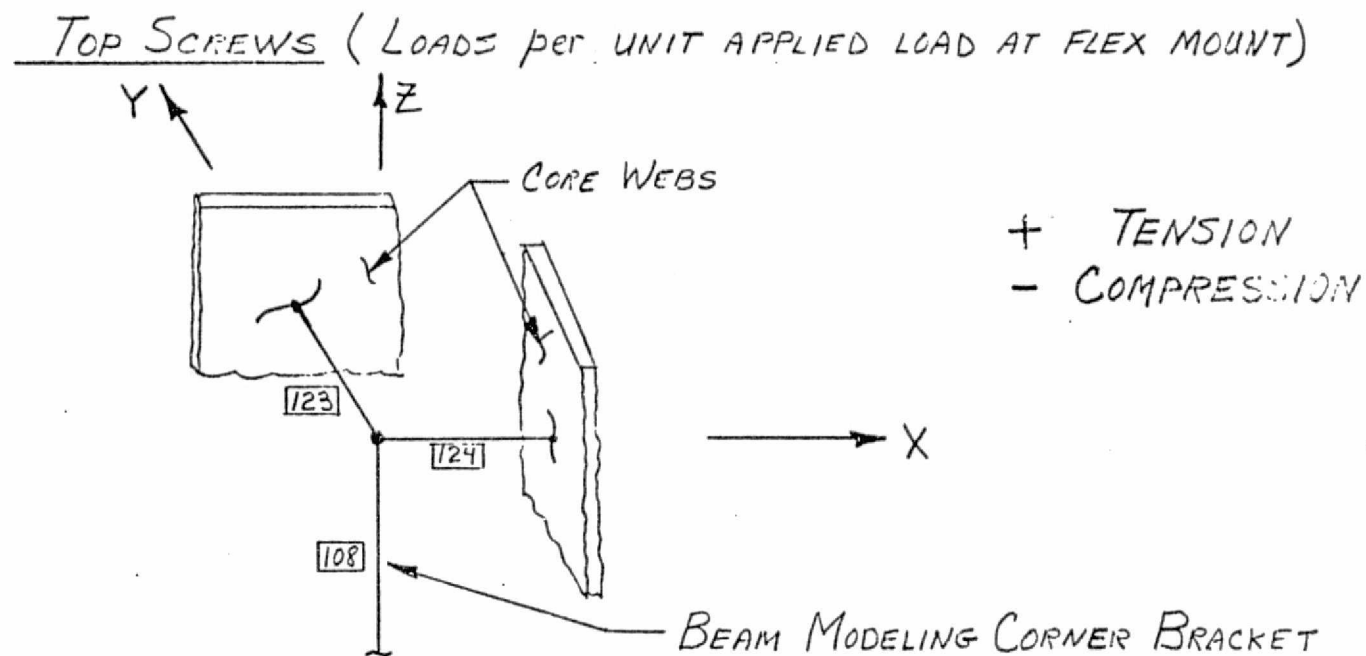
$F_{TY} = 19000$ PSI (MICROYIELD, 6061-T6 AL)

$$M.S. = \frac{19000}{(8060)(1.25)} - 1 = +.89$$

Project No. 16-1
August 1975
Asano Electronics
1780 Main
San Jose, California 95128

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CORNER BRACKET SCREW LOADS - TENSION



LOAD CONDITION: LATERAL X

$$\text{AXIAL FORCE } 123 = +.00720600$$

$$\text{AXIAL FORCE } 124 = +.00609596$$

LOAD CONDITION: LATERAL Y

$$\text{AXIAL FORCE } 123 = +.00497432$$

$$\text{AXIAL FORCE } 124 = +.00611891$$

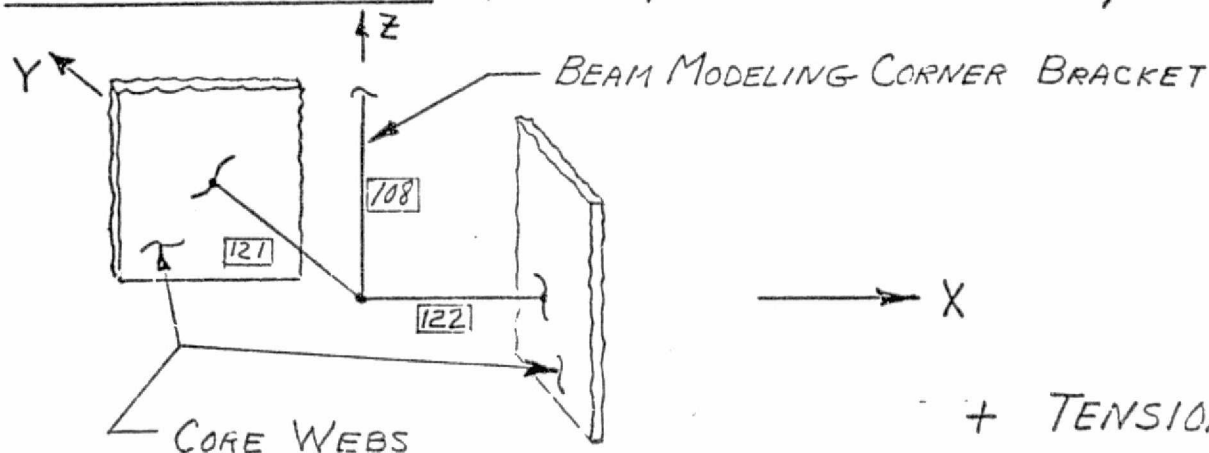
CORNER BRACKET SCREW LOADS-TENSION (CONTINUED)

LOAD CONDITION: AXIAL Z

$$\text{AXIAL FORCE 123} = -.00429666$$

$$\text{AXIAL FORCE 124} = -.00429666$$

BOTTOM SCREWS (LOADS per UNIT APPLIED LOAD)



+ TENSION
- COMPRESSION

LOAD CONDITION: LATERAL X

$$\text{AXIAL FORCE 121} = +.00299515$$

$$\text{AXIAL FORCE 122} = +.00077038$$

Asst. Engineer
Division
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CORNER BRACKET SCREW LOADS-TENSION (CONTINUED)

LOAD CONDITION : LATERAL Y

$$\text{AXIAL FORCE 121} = +.00142155$$

$$\text{AXIAL FORCE 122} = +.00364299$$

LOAD CONDITION : AXIAL Z

$$\text{AXIAL FORCE 121} = +.00259368$$

$$\text{AXIAL FORCE 122} = +.00259368$$

$$\text{APPLIED LOAD : LATERAL X} = 2500 \text{ LB}$$

$$\therefore \text{MAX. FORCE} = (2500)(.0072) = 18.0 \text{ LB}$$

$$\text{TENSILE LOAD CAPACITY} = 480 \text{ LB } \left(\frac{3}{16} \text{ STAINLESS STEEL} \right)$$

$$F_{TY} = 30000 \text{ PSI}$$

$$M.S. = \frac{480}{(18.0)(1.25)} - 1 = \text{AMPLE}$$

(NOT BASED ON MICROYIELD)

(OVER)

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Per No.

CORNER BRACKET SCREW LOADS - TENSION (CONTINUED)

BASED ON MICROYIELD:

$$F_{TY} = 30000 \text{ PSI}, \quad F_{SY} = (60\%)(30000) = 18000 \text{ PSI}$$

$$\therefore 60\% \text{ OF TENSILE CAPACITY} = (.60) 180 = 288 \text{ LB (YIELD)}$$

$$\text{USE } (.5)(288) = 144 \text{ LB (MICROYIELD)}$$

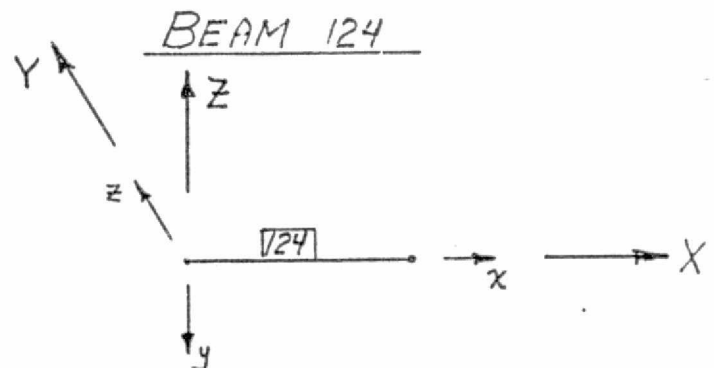
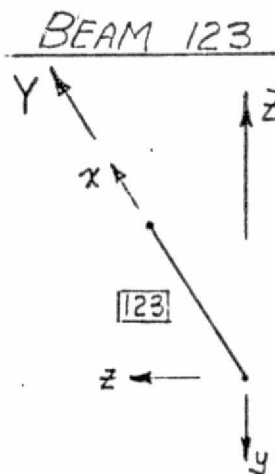
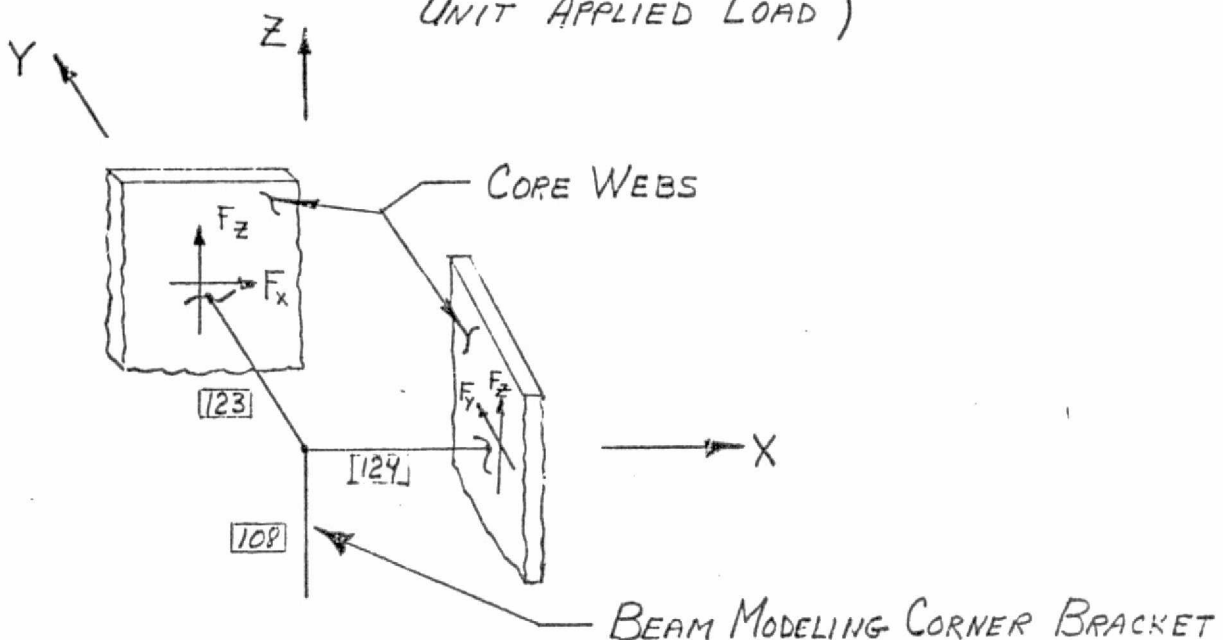
$$M.S. = \frac{144}{(18)(1.25)} - 1 = +5.40$$

Amco Electronics
1211 N. 1st St.
Tucson, Arizona 85701

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CORNER BRACKET - REQUIRED FRICTION LOADS

TOP SCREWS (SHEAR FORCES IN BEAMS 123, 124 per UNIT APPLIED LOAD)



CORNER BRACKET-REQUIRED FRICTION LOADS (CONTINUED)

LOAD CONDITION: LATERAL X

<u>BEAM NO.</u>	<u>LOCAL COORDINATE SYSTEM</u>	<u>GLOBAL COORDINATE SYSTEM</u>
123	$\begin{cases} F_y = -.0268320 \\ F_z = +.0217716 \end{cases}$	$\begin{aligned} F_x &= -.0217716 \\ F_z &= +.0268320 \end{aligned}$
124	$\begin{cases} F_y = -.0418328 \\ F_z = -.0215243 \end{cases}$	$\begin{aligned} F_y &= -.0215243 \\ F_z &= +.0418328 \end{aligned}$

LOAD CONDITION: LATERAL Y

123	$\begin{cases} F_y = -.0300679 \\ F_z = +.0179059 \end{cases}$	$\begin{aligned} F_x &= -.0179059 \\ F_z &= +.0300679 \end{aligned}$
124	$\begin{cases} F_y = -.0235791 \\ F_z = -.0181847 \end{cases}$	$\begin{aligned} F_y &= -.0181847 \\ F_z &= +.0235791 \end{aligned}$

LOAD CONDITION: AXIAL Z

123	$\begin{cases} F_y = +.0677596 \\ F_z = -.0118272 \end{cases}$	$\begin{aligned} F_x &= +.0118272 \\ F_z &= -.0677596 \end{aligned}$
124	$\begin{cases} F_y = +.0677596 \\ F_z = +.0118272 \end{cases}$	$\begin{aligned} F_y &= +.0118272 \\ F_z &= -.0677596 \end{aligned}$

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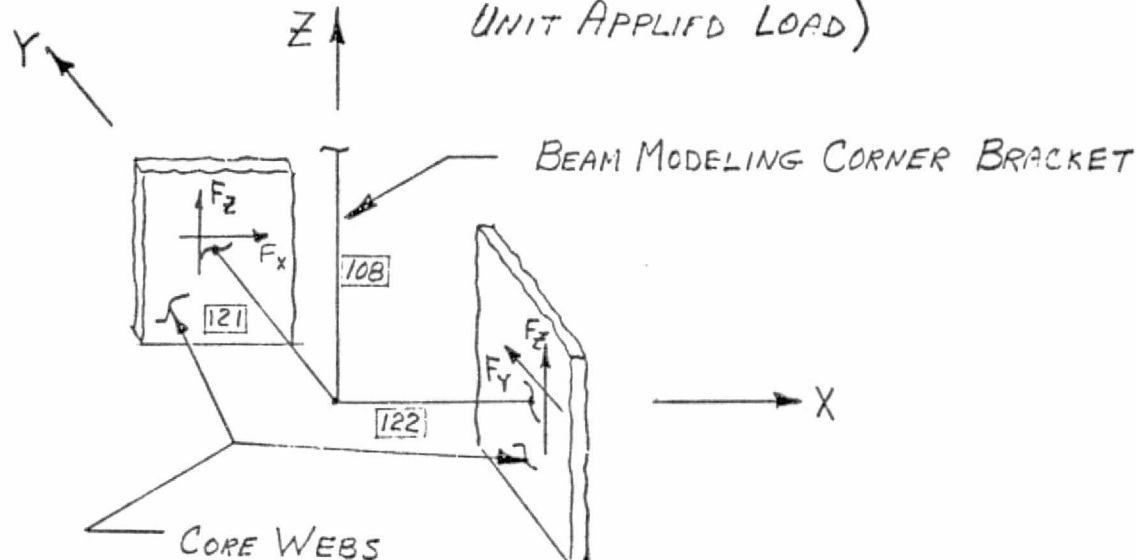
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CORNER BRACKET-REQUIRED FRICTION LOADS (CONTINUED)

RESULTANT SHEAR FORCES

	<u>BEAM 123</u>	<u>BEAM 124</u>
LOADING X-LAT.:	$F_s = .0345537$	$F_s = .0470455$
LOADING Y-LAT.:	$F_s = .0349957$	$F_s = .0297768$
LOADING Z-AXIAL:	$F_s = .0687841$	$F_s = .0687841$

BOTTOM SCREWS (SHEAR FORCES IN BEAMS 121, 122 per UNIT APPLIED LOAD)



BEAMS 121, 122 HAVE SAME LOCAL COORDINATE SYSTEM
AS BEAMS 123, 124

CORNER BRACKET-REQUIRED FRICTION LOADS (CONTINUED)

LOAD CONDITION : LATERAL X

BEAM NO.	LOCAL COORDINATE SYSTEM	GLOBAL COORDINATE SYSTEM
121	$\begin{cases} F_y = -.00193559 \\ F_z = +.00718612 \end{cases}$	$\begin{aligned} F_x &= -.00718612 \\ F_z &= +.00193559 \end{aligned}$
122	$\begin{cases} F_y = -.0550960 \\ F_z = -.00653728 \end{cases}$	$\begin{aligned} F_y &= -.00653728 \\ F_z &= +.0550960 \end{aligned}$

LOAD CONDITION : LATERAL Y

121	$\begin{cases} F_y = -.0443582 \\ F_z = +.00863778 \end{cases}$	$\begin{aligned} F_x &= -.00863778 \\ F_z &= +.0443582 \end{aligned}$
122	$\begin{cases} F_y = -.0000888 \\ F_z = -.00929113 \end{cases}$	$\begin{aligned} F_y &= -.00929113 \\ F_z &= +.0000888 \end{aligned}$

LOAD CONDITION : AXIAL Z

121	$\begin{cases} F_y = +.0501811 \\ F_z = +.00684454 \end{cases}$	$\begin{aligned} F_x &= -.00684454 \\ F_z &= -.0501811 \end{aligned}$
122	$\begin{cases} F_y = +.0501811 \\ F_z = -.00684454 \end{cases}$	$\begin{aligned} F_y &= -.00684454 \\ F_z &= -.0501811 \end{aligned}$

CORNER BRACKET-REQUIRED FRICTION LOADS (CONTINUED)

RESULTANT SHEAR FORCES

	<u>BEAM 121</u>	<u>BEAM 122</u>
LOADING X-LAT. :	$F_s = .0074422$	$F_s = .0554825$
LOADING Y-LAT. :	$F_s = .0451914$	$F_s = .0092916$
LOADING Z-AXIAL :	$F_s = .0506457$	$F_s = .0506457$

APPLIED LOAD X LATERAL = 2500 LB

$$\therefore \text{MAX. FORCE} = (2500)(.0555) = 138.8 \text{ LB}$$

TENSILE LOAD CAPACITY = 480 LB ($\frac{3}{16}$ STAINLESS STEEL)
 $F_{TY} = 30000 \text{ PSI}$

$$F_F = (.2)(480) = 96 \text{ LB}$$

$$M.S. = \frac{96}{(138.8)(1.25)} - 1 = -.45$$

$\therefore$ USE SERIES A-286 STAINLESS STEEL BOLTS

($F_{TY} = 85,000 \text{ PSI}$)

TENSILE LOAD CAPACITY = 1360 LB

17-6
SMM

7/3/75

CORNER BRACKET - REQUIRED FRICTION LOADS (CONTINUED)

$$\therefore F_F = (.2)(1360) = 272 \text{ LB}$$

$$M.S. = \frac{272}{(138.8)(1.25)} - 1 = +.57 \quad (\text{NOT BASED ON MICROYIELD})$$

BASED ON MICROYIELD:

$$F_{TY} = 85000 \text{ PSI} \Rightarrow F_{SY} = .60(85000) = 51000 \text{ PSI}$$

$$\therefore \text{USE } 60\% \text{ OF TENSILE CAPACITY} = .60(1350) \\ = 816 \text{ LB (SHEAR CAPACITY FOR YIELD)}$$

FOR MICROYIELD SHEAR CAPACITY, USE $.5(816) = 408 \text{ LB}$

$$\therefore F_F = (.2)(408) = 81.6 \text{ LB}$$

$$M.S. = \frac{81.6}{(138.8)(1.25)} - 1 = -.53$$

RCA

RECEIVED

IN 376

ALVIN W. SHEFFLER

MA 76-7-6

R. Packer

Location MS 67

Date 4 June 1976

From F. Hayes

Location MS 105

Telephone 2494

Subject SMM Flex Mount Assembly Stress Analysis

A detailed stress analysis was performed for the Solar Maximum Mission flex and fixed mount design for worst case static and dynamic loading conditions.

Items requiring attention are:

1. Tension in Flex Mount Bolts (5/16" Titanium)--Clamp Load required is 6800 lb (see p. 5-1.2).
2. Housing--Face Sheet Bolts (#10-32 Titanium)--Clamp Load required is 4370 lb (see p. 11-1.2, p. 14-1.4).
3. Corner Bracket Bolts (#10-32 Titanium)--Clamp Load required is 2000 lb (see p. 17-1.2).

To obtain these required clamp loads, the bolts may have to be solution treated. This decision should be made after further dynamic analysis of the panel.

Note that for the bolts mounting the housing to the face sheets, the e/d ratio must be 1.5 or greater to prevent tearout (e is the minimum distance from a hole center to the edge of the sheet and d is the diameter of the hole).

Also note that the flex arm was analyzed for a length (as defined on p. 4-1.1) of 2.6 inches. Thickness of the blade shall be .28 inches for instrument boxes C, E and .21 inches for all other boxes.

A summary of items analyzed and margins of safety is given on page 3. Material yield strength and load capabilities are shown on page 5. The detailed analysis of each item is presented in the appendix.

F. Hayes
F. Hayes

jal

dc W. Metzger
A. Sheffler ✓
G. Varadarajan

SOLAR MAX MISSION---FLEX MOUNT ASSEMBLY

Stress Analyses

1. Shear in Mounting Screw.
2. Bearing of Ball Bearing onto Ball Seat.
3. Bearing of Ball Seat onto Mounting Stud.
4. Bending in Flex Mount.
5. Tension in Flex Mount Bolts.
6. Shear in Flex Mount-to-Housing Dowels.
7. Bending in Flex Mount-to-Housing Dowels.
8. Bearing in Flex Mount (Dowel Pins).
9. Bearing in Housing (Dowel Pins).
10. Bending at Base of Housing.
11. Tension in Housing Mounting Screws.
12. Shear in Housing to ISP Dowel Pins.
13. Bearing in Housing (Dowel Pin).
14. Bearing in ISP Face Sheets (Dowel Pin).
15. Bending in ISP Face Sheets.
16. Corner Bracket Screw Loads--Tension.
17. Corner Bracket--Required Friction Loads.

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STRESS ANALYSIS

NO	TYPE LOADING/LOCATION	M.S.	CRITICAL LOAD CONDITION
1	SHEAR / MTG. STUD	+0.1	3.84g X PANEL BASE INPUT @ 39.Hz
2	BEARING / BALL SEAT	+0.41	..
3	BEARING / MTG. STUD	+1.00	..
4	BENDING / FLEX ARM	+0.09	..
5	TENSION / FLEX MOUNT BOLTS	+0.01 ⁽¹⁾	..
6	SHEAR / DOWEL PINS	+0.34	..
7	BENDING / DOWEL PINS	+0.60	..
8	BEARING / FLEX MOUNT	+0.28 ⁽²⁾	..
9	BEARING / HOUSING	+0.71 ⁽³⁾	..
10	BENDING / HOUSING	+0.24	..
11	TENSION / HOUSING BOLTS	AMPLE ⁽⁴⁾	..
12	SHEAR / DOWEL PINS	+0.09 ⁽⁴⁾	..
13	BEARING / HOUSING	+0.78 ⁽⁴⁾	..
14	BEARING / FACE SHEETS	+0.0 ⁽⁴⁾	..
15	BENDING / FACE SHEETS	+0.25	..
16	TENSION / CORNER BRKT. SCREWS	AMPLE	..
17	SHEAR / CORNER BRKT. SCREWS	+0.03	..

(1) REQUIRES CLAMP LOAD OF 6800 LB.

(2) INCLUDES FRICTION FORCE OF ONE ADJACENT BOLT

(3) TWO .. BOLTS

(4) REQUIRES BOLT CLAMP LOAD OF 4370 LB. → INCLUDES (2)

NOTE: • FRICTION COEFFICIENT OF .25 USED FOR ITEMS 13, 14
• ITEMS 5, 7, 11, 14, 16, 17 NOT BASED ON MICROYIELD



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	MATERIAL	SIZE	YIELD (PSI)	MICROYIELD (PSI)	TENSILE LOAD CAPACITY (LB)	SHEAR CAPACITY (LB)
(3)	Ti-6Al-4V	3/8		40000 54000 (shear)		
	TITANIUM	5/16 #10-32			6800 * 4370 *	
	STAINLESS STEEL DOWEL	5/16 ⁽²⁾ 3/16 1/8	(4) 60000 MIN	(4) 30000 MIN		5500 ⁽²⁾ 2000 900
(1)	2024-T4	-	50000 59000 (br)	25000 29500 (br)		
(3)	6061-T6			18000		

(1) per RCA STANDARDS MANUAL

(2) per MIL STD MS16555

(3) per NASA INSTRUCTION

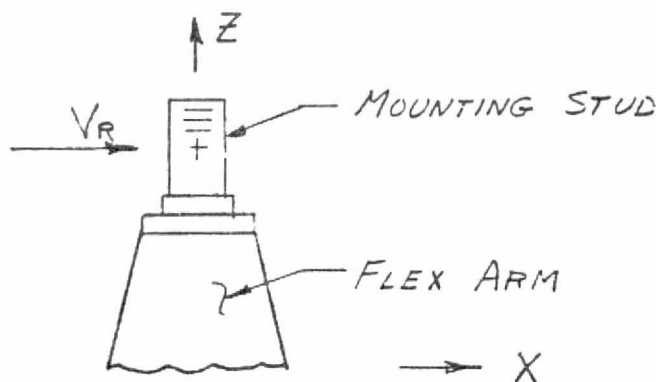
(4) per MTLs. SELECTOR GUIDE (PRODUCT DESIGN)

APPENDIX

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Prepared for: _____ Report No. 1-1.1
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1-1 SHEAR IN MOUNTING STUD



RESULTANT SHEAR FORCES IN THE MOUNTING STUD ARE OBTAINED DIRECTLY FROM THE APPLIED LOADS

Then,
$$\tau = \frac{V_R}{A_s}$$
 $A_s \equiv \text{AREA EFFECTIVE IN SHEAR}$

FROM THE MAXIMUM STATIC LOAD CASE - BOX 36-C2
 $V_R = 3870 \text{ LB.}$ $17.8g \text{ X} + 3.3g \text{ Y}$
(SMM coordinates)

FOR A $\frac{3}{8}$ -24 MOUNTING STUD W/ UNDERCUT .031"

$$A_s = \frac{\pi [(.375 - 2(.031))]^2}{4} = .0769 \text{ IN}^2$$

$$\therefore \tau = \frac{3870}{.0769} = 50,325 \text{ PSI}$$

FOR SHEAR YIELD, $F_{SY} = .60 F_{TY} = .60(90000) = 54,000 \text{ PSI}$

$$M.S. = \frac{54,000}{50,325} - 1 = \underline{\underline{+.07}}$$

Ti-6Al-4V
(BASED ON MICROYIELD)



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SHEAR IN MOUNTING STUD (CONT.)

FROM THE MAXIMUM DYNAMIC LOAD CASE - BOX 25-E2:

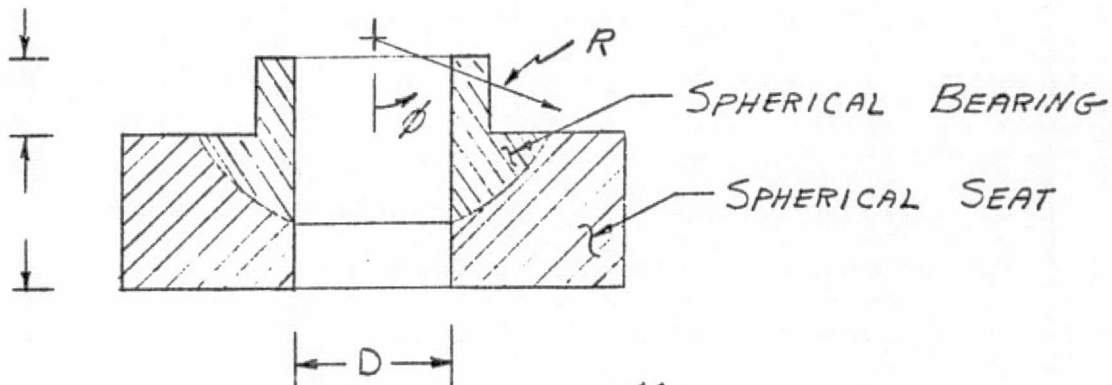
3.84g X PANEL BASE INPUT @ 39.Hz , $V_R = 4109. LB$

$$\therefore T = \frac{V_R}{A_s} = \frac{4109.}{.0769} = 53,435 \text{ psi}$$

AGAIN, FOR $F_{SY} = 54,000 \text{ psi}$

$$M.S. = \frac{54,000}{53,435} - 1 = \underline{\underline{+.01}}$$

2-1 BEARING OF BALL ONTO SPHERICAL SEAT



$$\text{CONTACT AREA} = D \left[\int_{30}^{55.2} R \sin \phi d\phi \right] = DR [-\cos \phi]_{30}^{55.2}$$

FOR $\frac{7}{8}$ " DIA. BALL

FOR 1" DIA BALL

$$D = .437"$$

$$R = .437"$$

$$A_c = .0564 \text{ IN}^2$$

$$A_c = .0564 \left(\frac{1}{7/8} \right) = .0645 \text{ IN}^2$$

Assume

- ONLY SHEAR FORCES RESULT IN DIRECT BEARING AGAINST BALLSEAT
- MOMENT IS CARRIED BY MOUNTING STUD (NEGLECT FRICTION FORCE BETWEEN BEARING AND SEAT)

$$\text{THEN, } S_{br} = \frac{V_R}{A_c}$$

FROM THE MAXIMUM STATIC CONDITION: $17.8g X + 3.3g Y$
(SMM coordinates)

$$V_R = 3870. \text{ LB}$$

$$\therefore S_{br} = \frac{3870}{.0645} = 60,040 \text{ psi}$$



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BEARING OF BALL ONTO SPHERICAL SEAT (CONT.)

For $F_{cy} = 90,000$ (Ti-6Al-4V MICROYIELD)

$$\therefore M.S. = \frac{90,000}{60,040} - 1 = \underline{\underline{+.50}}$$

FROM THE MAXIMUM DYNAMIC CONDITION:
3.84g X PANEL BASE INPUT @ 39. Hz

$$V_R = 4109. \text{ LB}$$

$$\therefore S_{br} = \frac{4109.}{.0645} - 1 = 63,750 \text{ psi}$$

AGAIN, FOR $F_{cy} = 90,000 \text{ psi}$

$$M.S. = \frac{90,000}{63,750} - 1 = \underline{\underline{+.41}}$$



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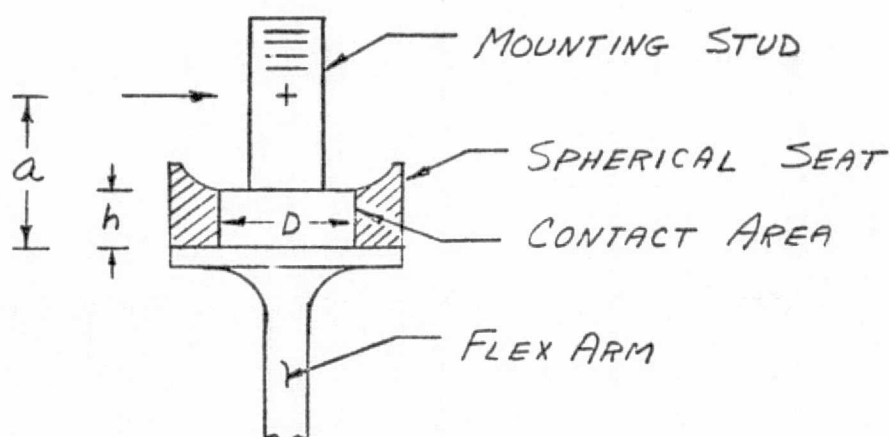
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3-1 BEARING OF SPHERICAL SEAT ONTO MOUNTING STUD



FOR $\frac{7}{8}$ " BALL

$$h = .183"$$

$$a = .562"$$

$$D = .437"$$

$$A_c = Dh = (.437)(.183) = .0800 \text{ IN}^2$$

$$\text{FOR } 1." \text{ BALL, } A_c \approx \frac{1}{7/8} (.0800) = .0914 \text{ IN}^2$$

Assume

- ONLY SHEAR FORCES RESULT IN DIRECT BEARING AGAINST BASE OF MOUNTING STUD
- MOMENT IS CARRIED BY MOUNTING STUD

$$\text{THEN, } S_{br} = \frac{V_R}{A_c}$$

$$\text{FOR STATIC LOADING: } 17.8g \text{ X} + 3.3g \text{ Y}$$

$$V_R = 3870. \text{ LB}$$

$$\therefore S_{br} = \frac{3870.}{.0914} = 42,350$$

$$\text{FOR } F_{cy} = 90,000 \text{ psi (Ti-6Al-4V Microyield)}$$

$$\text{M.S.} = \frac{90,000}{42,350} - 1 = \underline{\underline{+1.13}}$$



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BEARING OF BALLSEAT ONTO MOUNTING STUD (CONT.)

FOR DYNAMIC LOADING: 3.84 g PANEL BASE INPUT @ 39. Hz

$$V_R = 4109. \text{ LB}$$

$$\therefore S_{br} = \frac{4109.}{.0914} = 45,000 \text{ psi}$$

AGAIN, FOR $F_{CY} = 90,000 \text{ psi}$

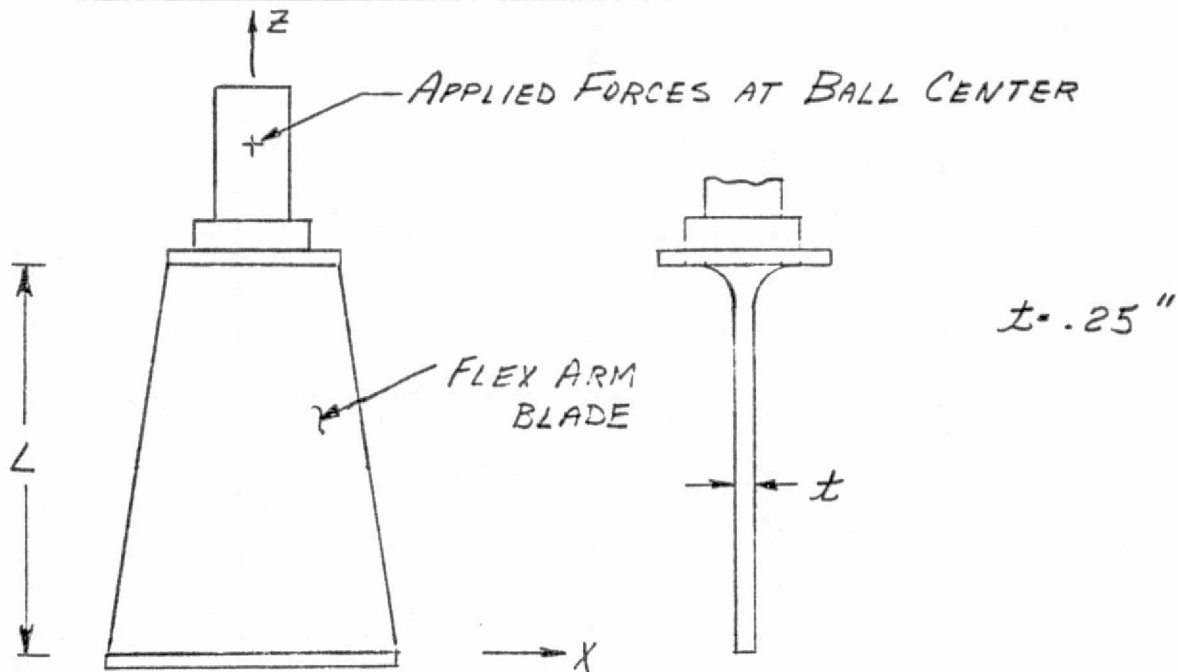
$$M.S. = \frac{90000}{45,000} - 1 = \underline{\underline{+1.00}}$$



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4-1 BENDING IN FLEX ARM



FROM THE DETAILED FINITE ELEMENT MODEL OF THE FLEX MOUNT, BENDING MOMENTS, SHEAR FORCES, AND AXIAL LOADS IN THE FLEX ARM ARE OBTAINED

THEN, STRESS IN THE FLEX ARM IS OBTAINED FROM

$$S = \sqrt{\sigma_N^2 + \tau_S^2} \quad \text{WITH} \quad \sigma_N = \frac{M_x c_z}{I_x} + \frac{M_y c_1}{I_y} + \frac{P}{A} \quad (\text{NORMAL})$$

$$\tau_S = \frac{V_R}{A_s} \quad (\text{SHEAR})$$

where

M = BENDING Moment
 V_R = Resultant Shear
 P = Axial Load
 A = Cross-Sectional Area
 A_s = Area Effective in shear
 c_1 = Distance to outer fiber - X dir
 c_2 = " " " " - Y dir



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BENDING IN FLEX ARM (CONT.)

FOR BLADE LENGTH, $L = 2.6''$

DYNAMIC LOAD CONDITION: $3.84g$ X PANEL BASE INPUT

BOX 36, ATTACH ARM C1 - FLEX ARM

$$P = 1900 \text{ LB} ; M_x = 460 \text{ IN LB} ; M_y = 1505 \text{ IN LB}$$

$$V_R = 1900 \text{ LB}$$

FOR $.28''$ THICK BLADE

$$S_N = \frac{1900}{.261} + \frac{460(.14)}{1.829 \times 10^{-3}} + \frac{(1505)(.484)}{.02035}$$

$$= 7280 + 35210 + 35800$$

$$S_N = 78,290 \text{ psi}$$

$$S_S = \frac{1900}{.261} = 7280 \text{ psi}$$

$$\therefore S_b = [(78290)^2 + (7280)^2]^{\frac{1}{2}} = 78,630 \text{ psi}$$

FOR $F_{TY} = 90,000 \text{ psi}$ (Ti-6Al-4V MICROYIELD)

$$M.S. = \frac{90,000}{78,630} - 1 = \underline{\underline{+.14}} \quad \leftarrow \begin{array}{l} \text{FLEX ARM} \\ \text{LARGE SIZE BLADE} \\ \text{(t = .28'')} \\ \text{BOXES C, E} \end{array}$$

FOR THE SAME LOAD CONDITION:

BOX 14, ATTACH ARM B1 - FLEX ARM



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BENDING IN FLEX ARM (CONT.)

$$P = 406 \text{ LB} ; M_x = 692. \text{ IN LB} ; M_y = 3313 \text{ IN LB}$$

$$V_R = 1400. \text{ LB}$$

For .21 " thick blade

$$S_N = \frac{406}{.378} + \frac{(692)(.105)}{1.359 \times 10^{-3}} + \frac{(3313)(.9)}{.1021}$$

$$= 1075 + 52,310 + 29,205$$

$$\therefore S_N = 82,590 \text{ psi}$$

$$S_s = \frac{1400}{.378} = 3,705 \text{ psi}$$

$$\therefore S_b = [(82,590)^2 + (3,705)^2]^{1/2} = 82,675 \text{ psi}$$

Again, for $F_{TV} = 90,000 \text{ psi}$

$$M.S. = \frac{90,000}{82,675} - 1 = \underline{\underline{+.09}}$$

FLEX ARM
MEDIUM SIZE BLADE
($t = .21$ ")
Boxes A, B, F, G, H

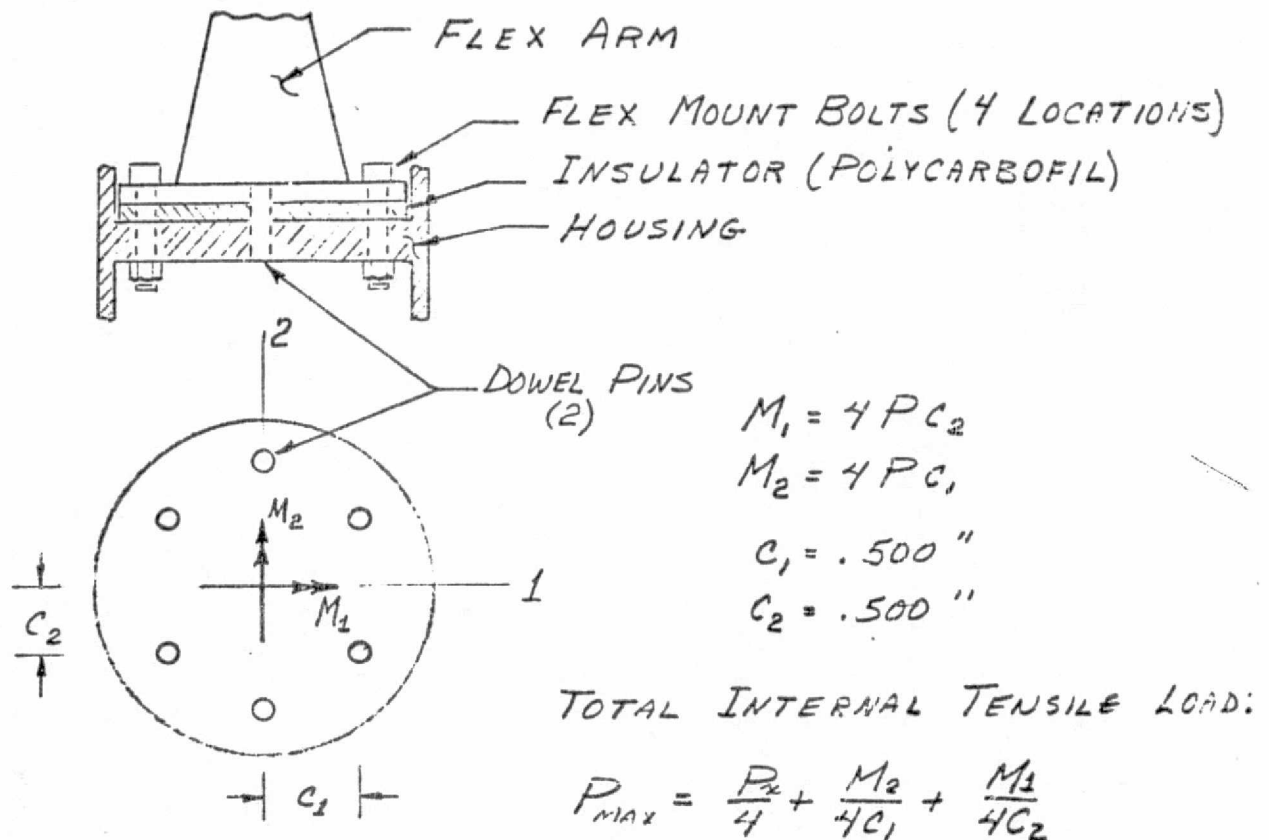


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5-1 TENSION IN FLEX MOUNT BOLTS



FOR STATIC LOADING: $17.8g X + 3.3g Y$

$$P_x = 3350. \text{ LB} ; M_2 = 9584. \text{ IN LB} ; M_1 = 2703. \text{ IN LB}$$

$$\therefore P = \frac{3350}{4} + \frac{9584}{4(.5)} + \frac{2703}{4(.5)} = 6981. \text{ LB}$$

HOWEVER, FOR A BLADE LENGTH OF 2.6" (OR LESS)

$$P_x = 3350. \text{ #} ; M_2 = 7729. \text{ " #} ; M_1 = 2180. \text{ " #}$$



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TENSION IN FLEX MOUNT BOLTS (CONT.)

∴ FOR STATIC LOADING,

$$P = \frac{3350}{4} + \frac{7729}{4(.5)} + \frac{(2180)}{4(.5)} = 5792 \text{ LB.}$$

FOR CLAMP LOAD CAPACITY = 6800 LB *

$$\therefore \text{M.S.} = \frac{6800}{5792} - 1 = \underline{\underline{+.17}}$$

FOR DYNAMIC LOADING: 3.84g PANEL BASE INPUT @ 39 Hz

$$P_x = 2625 \text{ LB}; M_2 = 4491 \text{ IN LB}; M_3 = 7692 \text{ IN LB}$$

(FOR BLADE LENGTH = 2.6")

$$\therefore P = \frac{2625}{4} + \frac{4491}{4(.5)} + \frac{7692}{4(.5)} = 6748 \text{ LB}$$

FOR CLAMP LOAD CAPACITY = 6800 LB

$$\text{M.S.} = \frac{6800}{6748} - 1 = \underline{\underline{+.01}}$$

* CLAMP LOAD NEEDED FOR 0. MARGIN IN DYNAMIC CONDITION



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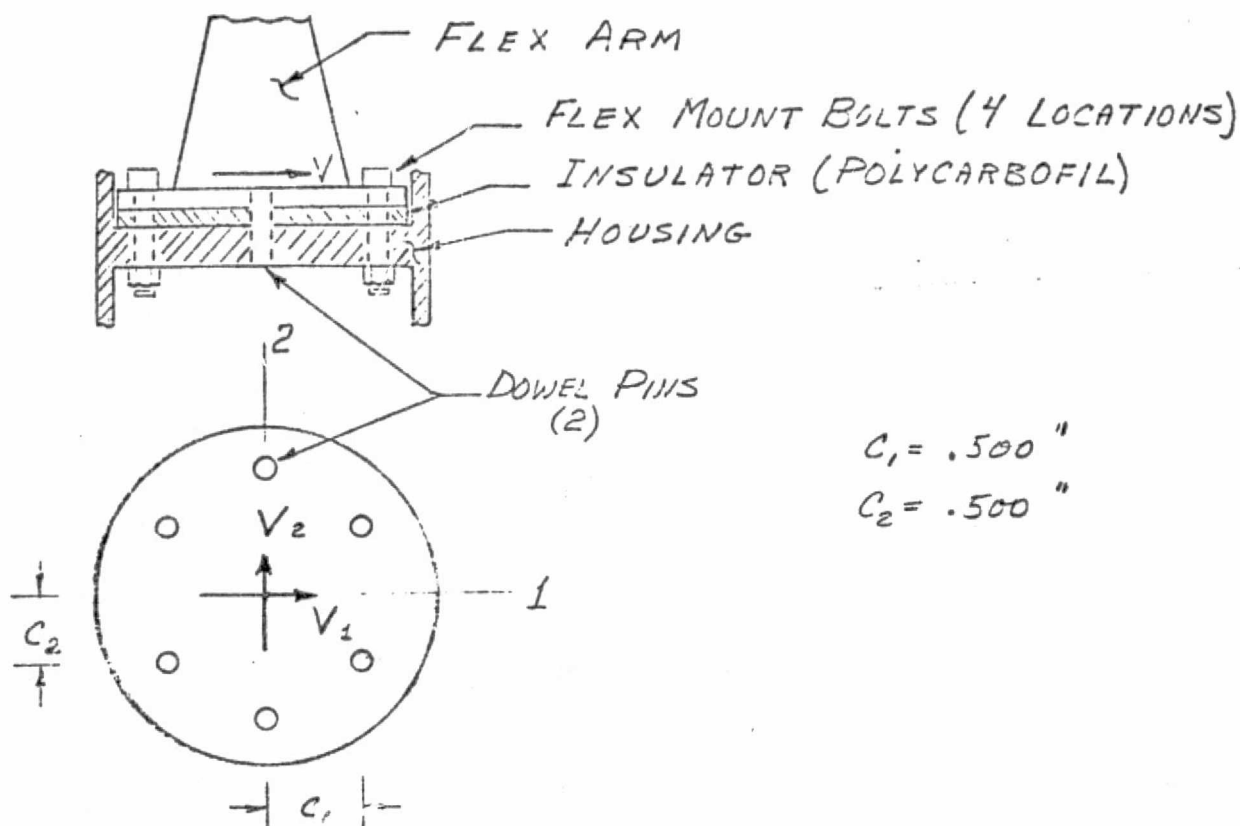
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6-1 SHEAR IN FLEX MOUNT-TO-HOUSING DOWELS



ASSUME:

- TOTAL SHEAR LOAD CARRIED BY SHEAR PINS
- BENDING MOMENT CARRIED BY FLEX MOUNT BOLTS

FOR STATIC LOADING: $17.8g X + 3.3g Y$

$$V_R = 3870. / 2 \text{ PINS} = 1935. \text{ LB}$$

MICROYIELD SHEAR CAPACITY FOR $\frac{5}{16}$ (A-286) Stainless Steel = $.50(5500) = 2750 \text{ LB.}$



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SHEAR IN FLEX MOUNT-TO-HOUSING DOWELS (CONT.)

$$\therefore M.S. = \frac{2750}{1935} - 1 = \underline{\underline{+.42}}$$

FOR DYNAMIC LOADING: 3.84g PANEL BASE INPUT @ 39. Hz

$$\therefore V_R = 4109 / 2 \text{ PINS} = 2055 \text{ LB.}$$

AGAIN, FOR 2750 LB MICROYIELD SHEAR CAPACITY

$$M.S. = \frac{2750}{2055} - 1 = \underline{\underline{+.34}}$$



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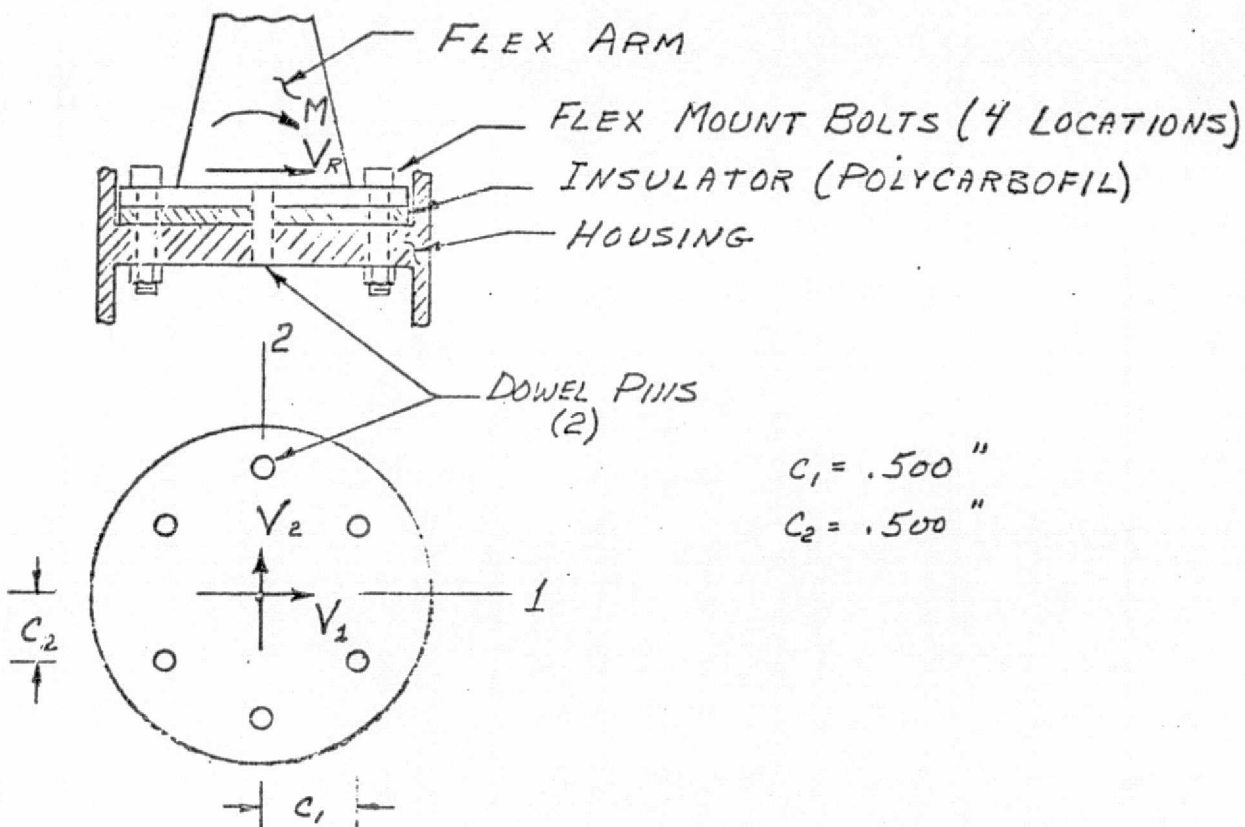
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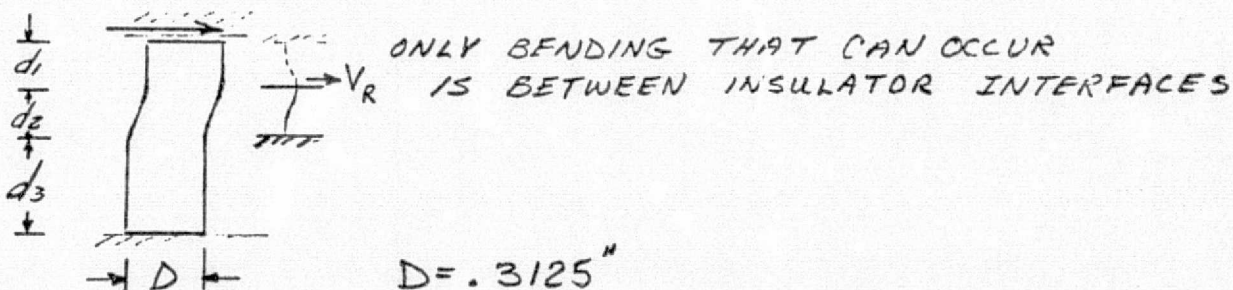
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7-1 BENDING IN FLEX MOUNT-TO-HOUSING DOWELS



ASSUME:

- BENDING MOMENT M CARRIED BY FLEX MOUNT BOLTS
- ONLY BENDING IS DUE TO SHEAR FORCE V_R (THIS IS CONSERVATIVE SINCE FOR A VERY SHORT LENGTH, DEFORMATION DUE TO SHEAR ONLY IS USUALLY ASSUMED).
- DEFORMATION PATTERN AS SHOWN



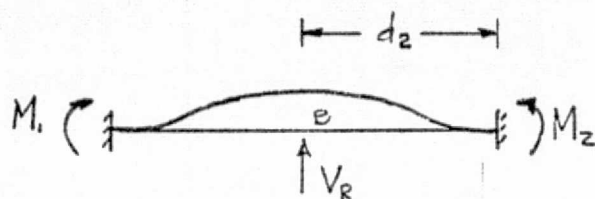


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BENDING IN FLEX. MOUNT-TO-HOUSING DOWELS (CONT.)

TO DETERMINE MAX BENDING MOMENT, CONSIDER
A FIXED-FIXED BEAM WITH CENTER LOAD V_R



$$M_{\max} = \frac{1}{8} V_R (2d_2)$$

$$d_2 = .218''$$

FOR STATIC LOADING: 17.8g X + 3.3g Y

$$V_R = 3870 / 2 \text{ PINS} = 1935 \text{ LB.}$$

$$\therefore M = \frac{1}{8} (1935)(2)(.218) = 105.5 \text{ IN LB}$$

SINCE $S_b = \frac{Mc}{I}$ where $c = \frac{.3125}{2} = .1563 \text{ IN}$

$$I = \pi \frac{(.3125)^4}{64} = 4.6813 \times 10^{-4} \text{ IN}^4$$

$$\therefore S_b = \frac{(105.5)(.1563)}{4.6813 \times 10^{-4}} = 35,225 \text{ psi}$$

FOR $F_{CY} = 60,000 \text{ MIN (STAINLESS STEEL)}$

$$M.S. = \frac{60,000}{35,225} - 1 = \underline{\underline{+.70}}$$

FOR DYNAMIC LOADING: 3.84g PANEL BASE INPUT

$$V_R = 4109 / 2 \text{ PINS} = 2055 \text{ LB}$$

$$\therefore M = \frac{1}{8} (2055)(2)(.218) = 112 \text{ IN LB}$$

$$\therefore S_b = \frac{Mc}{I} = \frac{(112)(.1563)}{4.6813 \times 10^{-4}} = 37,400 \text{ psi}$$



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BENDING IN FLEX MOUNT-TO-HOUSING DOWELS (CONT.)

SINCE $F_{CY} = 60,000 \text{ psi}$

$$M.S. = \frac{60,000}{37,400} - 1 = \underline{\underline{+0.60}}$$



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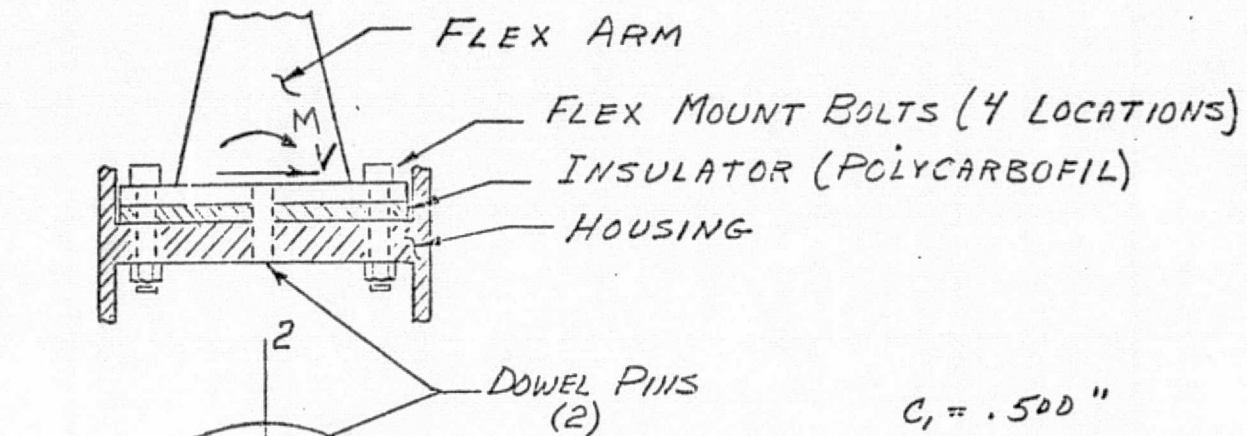
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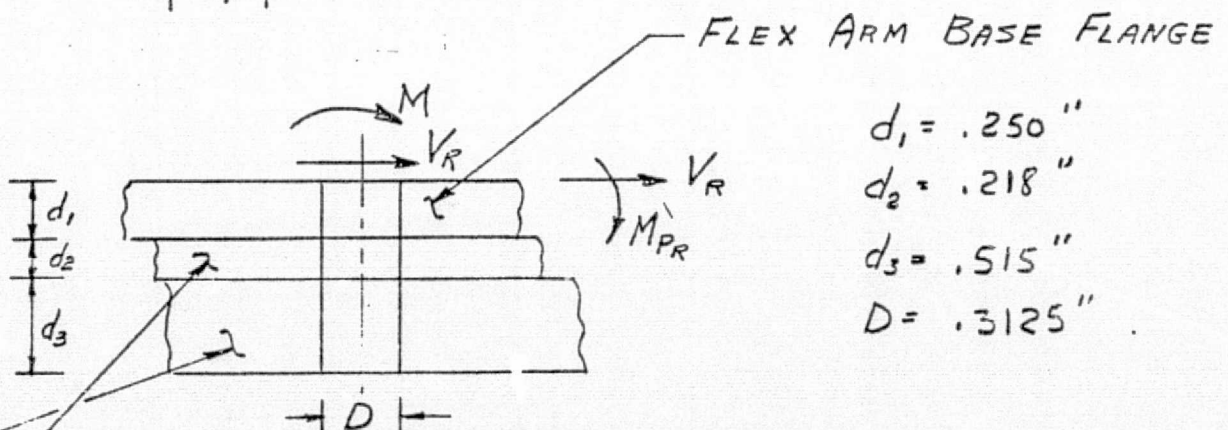
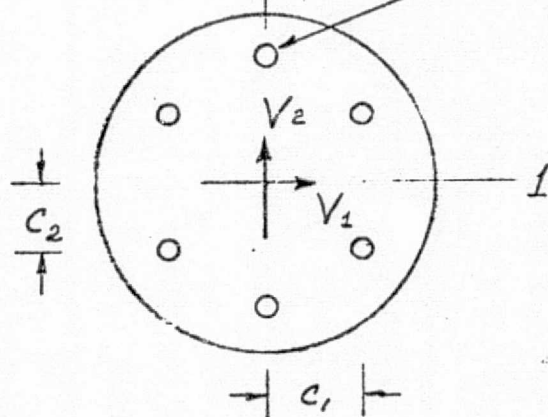
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8-1 BEARING IN FLEX MOUNT AT HOUSING INTERFACE



$$C_1 = .500''$$

$$C_2 = .500''$$



$$d_1 = .250''$$

$$d_2 = .218''$$

$$d_3 = .515''$$

$$D = .3125''$$

POLYCARBOFIL INSULATOR
HOUSING FLANGE

BEARING IN FLEX MOUNT AT HOUSING INTERFACE (CONT.)

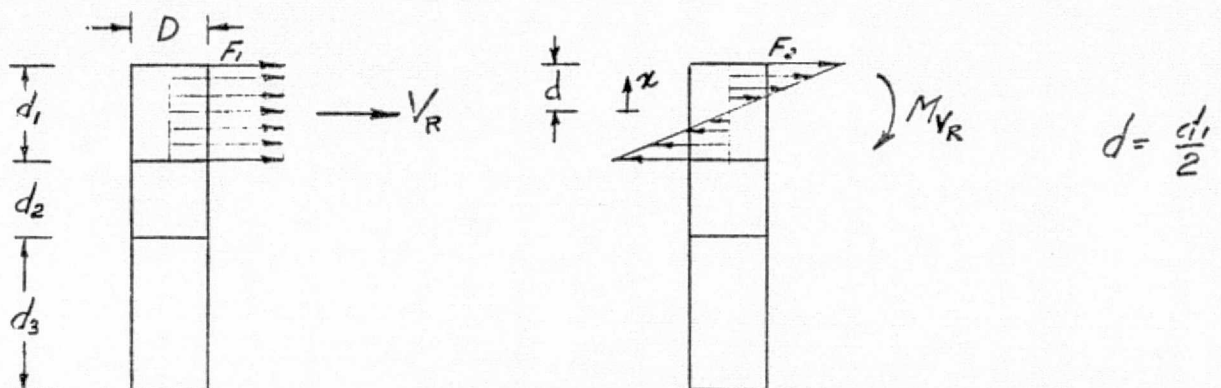
ASSUME:

- UNIFORM SHEAR LOAD P AND MOMENT M ACTING AS SHOWN
- MOMENT IS CARRIED BY FLEX MOUNT BOLTS (SECT. 5-1)

THE FORCE per UNIT WIDTH ACTING ON THE FLEX MOUNT CONSISTS OF:

- BEARING IN FLEX MOUNT DUE TO SHEAR FORCE V
- BEARING IN FLEX MOUNT DUE TO LOCAL BENDING MOMENT CAUSED BY SHEAR FORCE V_R (CONSERVATIVE ASSUMPTION)

THE DISTRIBUTION OF EACH IS:



$$\text{where } M_{VR} = 2 \int_0^d \left[F_2 \frac{x}{d} \right] (x) dx = 2 \frac{F_2}{d} \frac{x^3}{3} \Big|_0^d = 2 F_2 \frac{d^2}{3}$$

$$\text{Since } M_{VR} = V_R d, \quad F_2 = \frac{3 V_R}{2 d} \text{ or } F_2 = \frac{3 V_R}{d_1}$$

$$F_{\max} = F_1 + F_2 = \frac{V_R}{d_1} + \frac{3 V_R}{d_1} = \underline{\underline{\frac{4 V_R}{d_1}}}$$



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BEARING IN FLEX MOUNT AT HOUSING INTERFACE (CONT.)

NOW, CALCULATE BEARING STRESS BASED ON
BEARING CONTACT AREA

$$A_c = D d_i$$

$$\therefore S_{br} = \frac{F}{D} = \frac{4 V_R}{d_i D}$$

FOR STATIC LOADING: $17.8g X + 3.3g Y$

$$V_R = \frac{3870}{2 \text{ PINS}} = 1935.$$

$$\therefore S_{br} = \frac{4(1935.)}{(25)(.3125)} = 99,075 \text{ psi}$$

ASSUME PART OF LOAD IS CARRIED BY $\frac{5}{16}$ " Ti bolt
FRICTION FORCE: $F_f = (.2)(6800)^* = 1360. \text{ LB}$

$$\therefore V_R = \frac{3870 - 1360}{2} = 1255. \text{ LB}$$

$$\therefore S_{br} = \frac{4(1255.)}{(25)(.3125)} = 64,260 \text{ psi}$$

FOR $F_{cr} = 90,000 \text{ psi}$ (Ti-6Al-4V Microyield)

$$\therefore M.S. = \frac{90,000}{64,260} - 1 = \underline{\underline{+.40}}$$

FOR DYNAMIC LOADING: $3.84g X$ PANEL BASE INPUT @ $37. \text{ Hz}$

$$V_R = \frac{4109.}{2 \text{ PINS}} = 2055. \text{ LB}$$

* CLAMP FORCE NEEDED FOR 0. MARGIN IN BOLT TENSION (SECT. 5-1)



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BEARING IN FLEX MOUNT AT HOUSING INTERFACE (CONT.)

FOR ONE $\frac{5}{16}$ " BOLT FRICTION FORCE EFFECTIVE

$$V_R = \frac{4109. - 1360}{2} = 1375. \text{ LB}$$

$$\therefore S_{br} = \frac{4(1375)}{(.25)(.3125)} = 70,400 \text{ psi}$$

AGAIN, FOR $F_{CY} = 90,000 \text{ psi}$

$$M.S. = \frac{90,000}{70,400} - 1 = \underline{\underline{+.28}}$$



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BEARING IN HOUSING AT FLEX ARM BASE (CONT.)

$$\text{SINCE } F_{MAX} = F_1 + F_2 = \frac{4V_R}{d_3} \quad \text{and} \quad S_{br} = \frac{F}{D}$$

$$\therefore S_{br} = \frac{4V_R}{d_3 D} \quad (\text{See p. 8.1.3})$$

FOR STATIC LOADING: 17.8 g X + 3.3 g Y

$V_R = 1255 \text{ LB}$ (ASSUMING 1 BOLT FRICTION FORCE EFFECTIVE) - See SECT 8-1

$$S_{br} = \frac{4(1255)}{(0.515)(0.3125)} = 31,200 \text{ psi}$$

FOR BEARING MICROYIELD (2024 T4)

$$F_{brMY} = \frac{1}{2} F_{brY} = \frac{1}{2} (59,000) = 29,500 \text{ psi}$$

$$\therefore M.S. = \frac{29,500}{31,200} - 1 = -0.05$$

ASSUMING 2 OF THE 4 $\frac{5}{16}$ " Ti bolts ARE EFFECTIVE, $F_f = 2(2)(6850) = 2720 \text{ LB}$

$$\therefore V_R = \left(\frac{3870 - 2720}{2} \right) = 575 \text{ LB}$$

$$\therefore S_{br} = \frac{4(575)}{(0.515)(0.3125)} = 14,300 \text{ psi}$$

$$\therefore M.S. = \frac{29,500}{14,300} - 1 = \underline{\underline{+1.06}}$$



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BEARING IN HOUSING AT FLEX ARM BASE (CONT.)

FOR DYNAMIC LOADING: $3.84g \times$ PANEL BASE INPUT @ $39. Hz$

$$V_R = \frac{4109 - 2720}{2} = 695. \text{ LB (ASSUMING } 2 \frac{5}{16}''$$

BOLTS FRICTION FORCE IS EFFECTIVE)

$$\therefore S_{br} = \frac{4(695)}{(.515)(.3125)} = 17,300 \text{ psi}$$

AGAIN, FOR $F_{brMY} = 29,500 \text{ psi}$

$$M.S. = \frac{29,500}{17,300} - 1 = \underline{\underline{+.71}}$$



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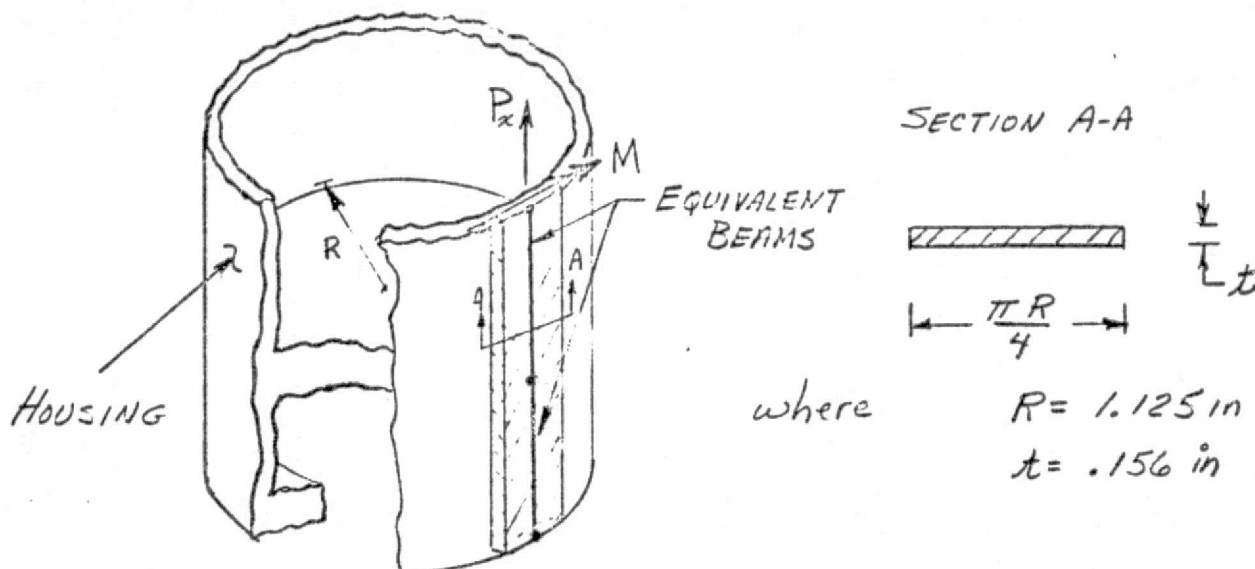
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10-1 BENDING STRESS IN HOUSING



where

$$R = 1.125 \text{ in}$$

$$t = .156 \text{ in}$$

SINCE, IN THE DETAILED FINITE ELEMENT MODEL, THE HOUSING IS MODELED AS EQUIVALENT BEAMS, THE STRESS IN THE HOUSING CAN BE OBTAINED BY

$$S_b = \frac{P_x}{A} + \frac{Mc}{I}$$

where

$$A = .138 \text{ in}^2$$

$$I = 2.797 \times 10^{-4} \text{ in}^4$$

$$c = .078 \text{ in}$$

FOR STATIC LOADING: $17.8g \text{ X} + 3.3g \text{ Y}$

$$P_x = 1200. \text{ LB}, \quad M = 32.5 \text{ in. LB.}$$

$$\therefore S_b = \frac{1200}{.138} + \frac{(32.5)(.078)}{2.797 \times 10^{-4}} = 17,760 \text{ psi}$$



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BENDING STRESS IN HOUSING (CONT.)

FOR $F_{cy} = 25,000 \text{ psi}$ (2024 T4 MICROYIELD)

$$M.S. = \frac{25,000}{17,760} - 1 = \underline{\underline{+.41}}$$

FOR DYNAMIC LOADING: 3.84g X PANEL BASE INPUT @ 32 Hz

$$P_z = 1278. \text{ LB} , \quad M = 38.8 \text{ IN LB}$$

$$\therefore S_b = \frac{1278}{.138} + \frac{(38.8)(.078)}{2.797 \times 10^{-4}} = 20,100 \text{ psi}$$

AGAIN, FOR $F_{cy} = 25,000 \text{ psi}$ (BASED ON MICROYIELD)

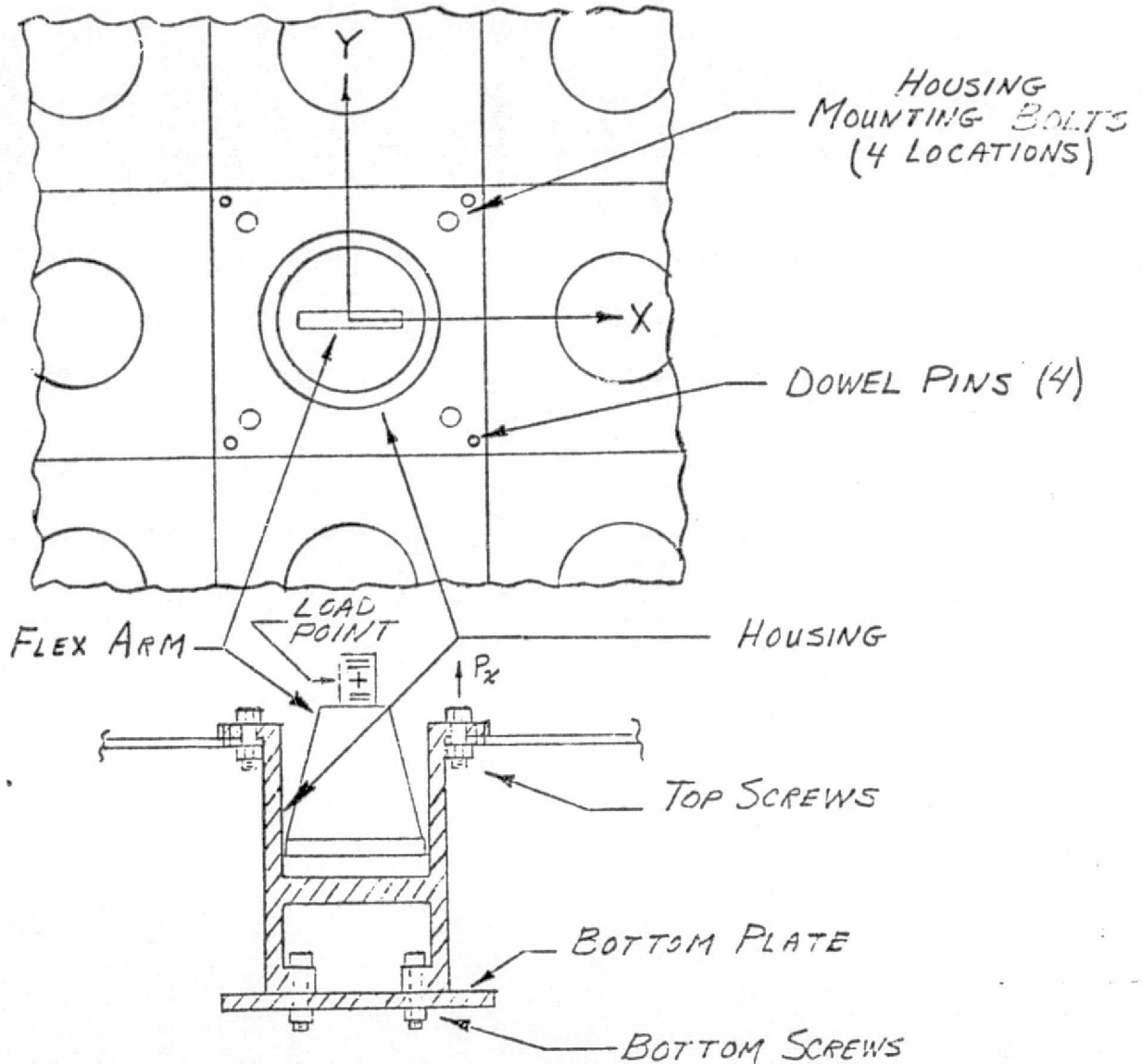
$$M.S. = \frac{25,000}{20,100} - 1 = \underline{\underline{+.24}}$$



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11-1 TENSION IN HOUSING MOUNTING BOLTS



NOTE: THE LOAD IN THE TOP SCREWS IS GREATER THAN THE LOAD IN THE BOTTOM SCREWS.



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TENSION IN HOUSING MOUNTING BOLTS (CONT.)

FOR STATIC LOADING: $17.8g X + 3.3g Y$

$$P_x = 710. \text{ LB}$$

FOR 10-32 MOUNTING BOLTS WITH $F_y = 150,000 \text{ psi}$

$$\text{CLAMP LOAD} = 4370 \text{ LB.}^*$$

$$\therefore \text{M.S.} = \frac{4370}{710} - 1 = \underline{\underline{\text{AMPLE}}}$$

FOR DYNAMIC LOADING: $3.84g X$ PANEL BASE INPUT TO
BOX 25-E1 @ $39. \text{ Hz}$

$$P_x = 737. \text{ LB}$$

FOR CLAMP LOAD = 4370 LB.

$$\text{M.S.} = \frac{4370}{737} - 1 = \underline{\underline{\text{AMPLE}}}$$

* CLAMP LOAD REQUIRED FOR BEARING IN FACE SHEET (SECT. 14-1)



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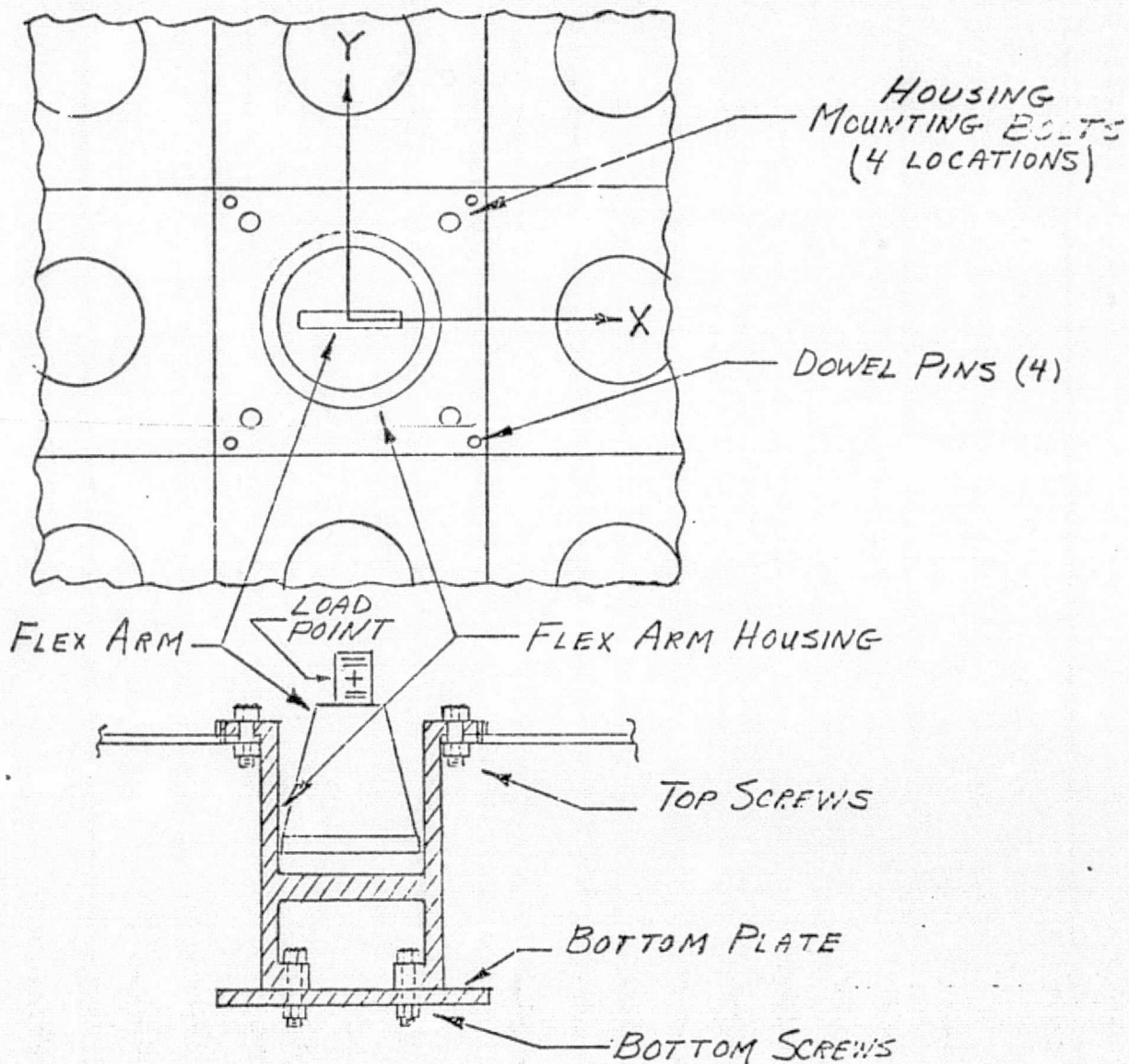
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12-1 SHEAR IN HOUSING-TO-ISP DOWEL PINS



FOR STATIC LOADING: $17.8g X + 3.3g Y$

$V_R = 1211.4B$



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SHEAR IN HOUSING-TO-ISP DOWEL PINS (CONT.)

FOR $\frac{1}{8}$ " STAINLESS STEEL (A-286) DOWELS

$$\text{SHEAR CAPACITY} = 900. \text{ LB}$$

$$\therefore \text{M.S.} = \frac{900}{1211} - 1 = -.26$$

CONSIDERING FRICTION FORCE $F_f = .2(4370)^* = 874. \text{ LB}$
TO BE EFFECTIVE

$$V_R = 1211. - 874 = 337. \text{ LB}$$

$$\therefore \text{M.S.} = \frac{900}{337} - 1 = \underline{\underline{+1.67}}$$

FOR DYNAMIC LOADING: 3.84g X PANEL BASE INPUT @ 37. Hz

$$V_R = 1286. - 874 = 412. \text{ LB (FRICTION INCLUDED)}$$

AGAIN, FOR SHEAR CAPACITY = 900. LB

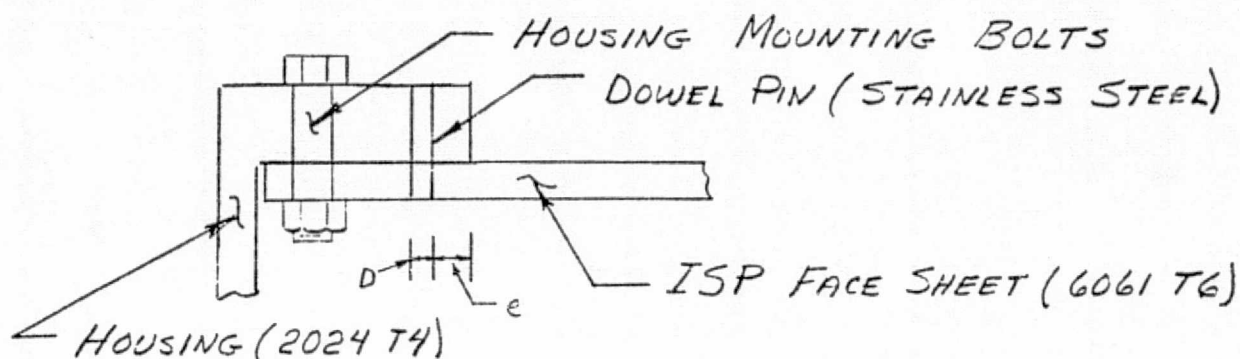
$$\text{M.S.} = \frac{900}{412} - 1 = \underline{\underline{+1.18}}$$

BASED ON MICROYIELD, SHEAR CAPACITY = 450 LB

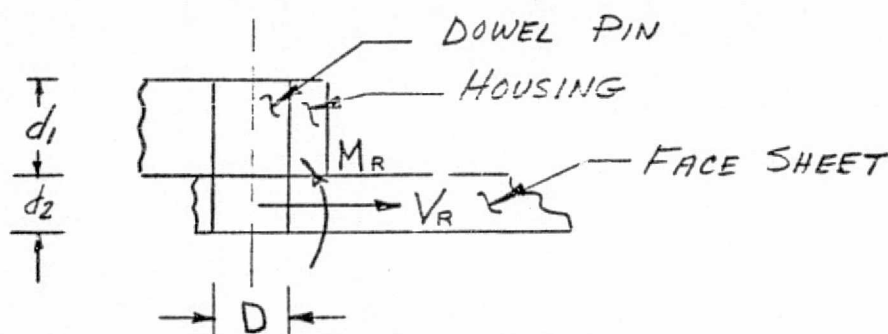
$$\therefore \text{M.S.} = \frac{450}{412} - 1 = \underline{\underline{+.09}}$$

* CLAMP LOAD REQUIRED FOR BEARING IN FACE SHEET (SECT. 14-1)

13-1 BEARING IN HOUSING AT ISP INTERFACE



FROM THE DETAILED FINITE ELEMENT MODEL OF THE FLEX MOUNT, LOADS ARE OBTAINED AS SHOWN



$$d_1 = .187''$$

$$d_2 = .125''$$

$$D = .125''$$

ASSUME:

- UNIFORM SHEAR PLUS BENDING MOMENT ACTING AS SHOWN
- MOMENT IS CARRIED BY TENSION AND COMPRESSION IN HOUSING MOUNTING BOLTS (SECTION 11-1)

THE TOTAL FORCE per UNIT WIDTH ACTING ON THE HOUSING CONSISTS OF:

- BEARING IN HOUSING DUE TO SHEAR FORCE V_R
- BEARING IN HOUSING DUE TO EQUIVALENT BENDING MOMENT CAUSED BY SHEAR FORCE V_R



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BEARING IN HOUSING AT ISP INTERFACE (CONT.)

IGNORING LOCAL BENDING,

$$S_{br} = \frac{V_R}{d_1 D}$$

FOR STATIC LOADING: $17.8g X + 3.3g Y$

$V_R = 1211. LB$ (NEGLECTING FRICTION)

$$\therefore S_{br} = \frac{1211}{(.187)(.125)} = 51,810 \text{ psi}$$

FOR $F_{brY} = 29,500 \text{ psi}$ (2024 T4 MICROYIELD) *

$$M.S. = \frac{29,500}{51,810} - 1 = -.43$$

NOT BASED ON MICROYIELD, $F_{brY} = 59,000 \text{ psi}$

$$\therefore M.S. = \frac{59,000}{51,810} - 1 = \underline{+.14}$$

FOR DYNAMIC LOADING: $3.84g X$ PANEL BASE INPUT @ 39. Hz

$V_R = 1286. LB$ (NEGLECTING FRICTION)

$$\therefore S_{br} = \frac{1286}{(.187)(.125)} = 55,020 \text{ psi}$$

NOT BASED ON MICROYIELD, $F_{brY} = 59,000 \text{ psi}$

$$\therefore M.S. = \frac{59,000}{55,020} - 1 = \underline{+.07}$$

INCLUDING LOCAL BENDING IN THE ANALYSIS, WE HAVE
THE FOLLOWING LOAD DISTRIBUTIONS OCCURRING;

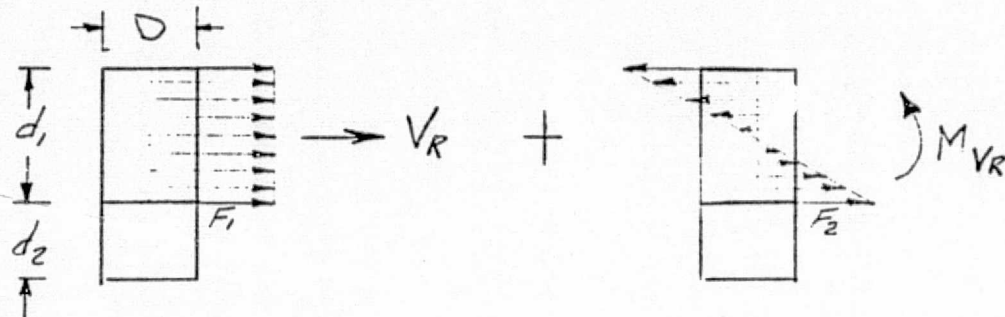
* NOTE: $\frac{e}{d}$ RATIO MUST BE 1.5 OR GREATER TO PREVENT TEAROUT



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BEARING IN HOUSING AT ISP INTERFACE (CONT.)



THUS, $F_{MAX} = F_1 + F_2 = \frac{V_R}{d_1} + \frac{3V_R}{d_1} = 4 \frac{V_R}{d_1}$

SINCE $S_{br} = \frac{F}{D}$, THEN $S_{br} = \frac{4V_R}{d_1 D}$

FOR STATIC LOADING: $17.8g \bar{X} + 3.3g \bar{Y}$

$V_R = 731 \text{ LB}$ (FRICTION OF 1 #10 BOLT INCLUDES)
CLAMP LOAD = 2400 LB

$\therefore S_{br} = \frac{4(731)}{(0.187)(.125)} = 125,100 \text{ psi}$

FOR $F_{br} = 29,500 \text{ psi}$ (2024 T4 MICROYIELD)*

M.S. = $\frac{29,500}{125,100} - 1 = -.76$

FOR A CLAMP LOAD OF 4800 LB, $F_f = .2(4800) = 960 \text{ LB}$

$\therefore V_R = 1131 - 960 = 171 \text{ LB}$ (SEC 12-1.1)

$\therefore S_{br} = \frac{4(171)}{(0.187)(.125)} = 29,300 \text{ psi}$

* $\frac{e}{d}$ ratio must be 1.5 or greater to prevent tearout



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BEARING IN HOUSING AT ISP INTERFACE (CONT.)

For $F_{cr} = 29,500 \text{ psi}$ (2024 T4 MICROYIELD)

$$M.S. = \frac{29,500}{29,300} - 1 = \underline{\underline{+.01}} \quad \leftarrow \text{NEED CLAMP LOAD OF 4800 LB}$$

FOR DYNAMIC LOADING: $3.84g \times \text{PANEL BASE INPUT @ } 39. \text{ Hz}$

$$V_R = 806. \text{ LB}$$

$$\therefore S_{br} = \frac{4(806)}{(.187)(.125)} = 137,925 \text{ psi}$$

For $F_{br} = 29,500 \text{ psi}$

$$M.S. = \frac{29,500}{137,925} - 1 = -.79$$

FOR A CLAMP LOAD OF 5570 LB., $F_f = .2(5570) = 1114. \text{ LB}$

$$V_R = 1286. - 1114 = 172. \text{ LB}$$

$$\therefore S_{br} = \frac{4(172)}{(.187)(.125)} = 29,435. \text{ psi}$$

FOR, $F_{br Y} = 29,500 \text{ psi}$

$$M.S. = \frac{29,500}{29,435} - 1 = \underline{\underline{+.00}} \quad \leftarrow \text{NEED CLAMP LOAD OF 5570 LB}$$

NOW, IF WE USE A FRICTION COEFFICIENT OF .25 and
BASE MARGINS ON YIELD INSTEAD OF MICROYIELD,
(SEE SECT. 14-1)



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BEARING IN HOUSING AT ISP INTERFACE (CONT.)

$$F_F = .25(4370)^* = 1092. \text{ LB}$$

FOR STATIC LOADING, $V_R = 1211 - 1092 = 119.$

$$\therefore S_{br} = \frac{4(119)}{(.187)(.125)} = 20,400 \text{ psi}$$

FOR $F_{BRY} = 59,000 \text{ psi}$ (NOT BASED ON MICROFIELD)

$$M.S. = \frac{59,000}{20,400} - 1 = \underline{\underline{+.90}} \leftarrow \begin{array}{l} \text{CLAMP LOAD} \\ \text{OF } 4370 \text{ LB} \\ (\mu = .25) \end{array}$$

FOR DYNAMIC LOADING, $V_R = 1286 - 1092 = 194.$

$$\therefore S_{br} = \frac{4(194)}{(.187)(.125)} = 33,200 \text{ psi}$$

FOR $F_{ERY} = 59,000 \text{ psi}$

$$M.S. = \frac{59,000}{33,200} - 1 = \underline{\underline{+.78}} \leftarrow \begin{array}{l} \text{CLAMP LOAD} \\ \text{OF } 4370 \text{ LB} \\ (\mu = .25) \end{array}$$

* CLAMP LOAD USED FOR 0. MARGIN IN FACE SHEET (14-1.4)

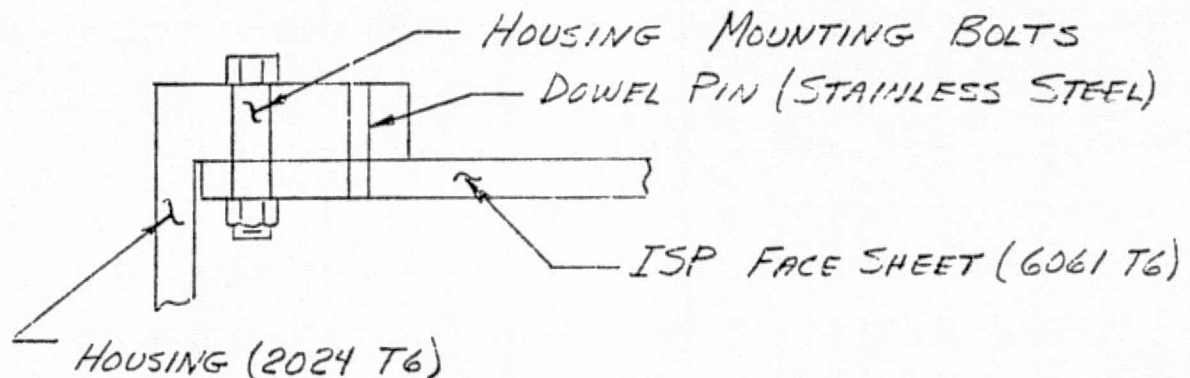


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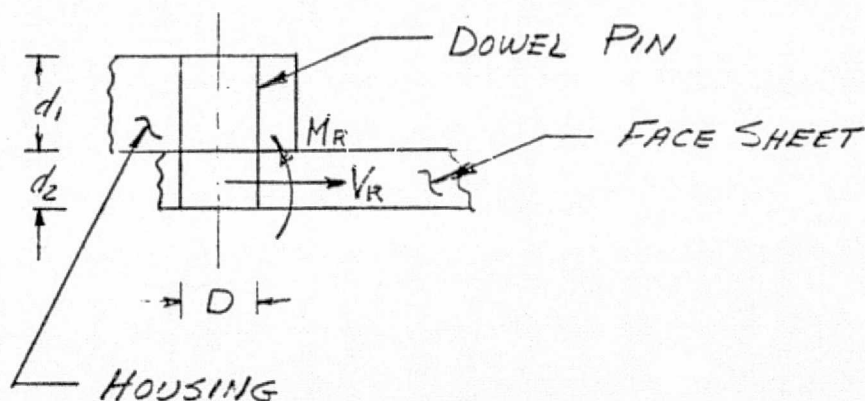
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14-1 BEARING IN ISP FACE SHEETS



FROM THE DETAILED FINITE ELEMENT MODEL OF THE FLEX MOUNT, LOADS ARE OBTAINED AS SHOWN



ASSUME:

- UNIFORM SHEAR PLUS BENDING MOMENT ACTING AS SHOWN
- MOMENT IS CARRIED BY TENSION AND COMPRESSION IN HOUSING MOUNTING BOLTS (SECTION 11-1)

THE TOTAL FORCE per UNIT WIDTH ACTING ON THE FACE SHEET CONSISTS OF:

- BEARING IN FACE SHEET DUE TO SHEAR FORCE V_R
- BEARING IN FACE SHEET DUE TO EQUIVALENT BENDING MOMENT CAUSED BY SHEAR FORCE V_R



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BEARING IN ISP FACE SHEETS (CONT.)

IGNORING LOCAL BENDING,

$$S_{br} = \frac{V_R}{d_2 D}$$

FOR STATIC LOADING: $12.8g X + 3.3g Y$

$V_R = 1211. \text{ LB}$ (NEGLECTING FRICTION)

$$\therefore S_{br} = \frac{1211.}{(.125)(.125)} = 77,500 \text{ psi}$$

FOR $F_{BRY} = 25,000 \text{ psi}$ (6061 T6 MICROYIELD)

$$M.S. = \frac{25,000}{77,500} - 1 = -.68$$

NOT BASED ON MICROYIELD, $F_{BRY} = 50,000 \text{ psi}$

$$\therefore M.S. = \frac{50,000}{77,500} - 1 = -.35$$

FOR DYNAMIC LOADING: $3.84g X$ PANEL BASE INPUT

$V_R = 1286. \text{ LB}$ (NEGLECTING FRICTION)

$$\therefore S_{br} = \frac{1286.}{(.125)(.125)} = 82,300 \text{ psi}$$

FOR $F_{BRY} = 50,000 \text{ psi}$ (NOT BASED ON MICROYIELD)

$$M.S. = \frac{50,000}{82,300} - 1 = -.39$$

INCLUDING THE EFFECT OF FRICTION FORCE ON

BEARING, $F_f = \mu N = (.2)(2525)^* = 505. \text{ LB}$

* CLAMP LOAD REQUIRED FOR 0. SAFETY MARGIN



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BEARING IN ISP FACE SHEETS (CONT.)

FOR STATIC LOADING, $V_R = 1211 - 505 = 706$ LB

$$\therefore S_{br} = \frac{706}{(.125)(.125)} = 45,200 \text{ psi}$$

FOR $F_{BRY} = 50,000 \text{ psi}$ (NOT BASED ON MICROYIELD)

$$M.S. = \frac{50,000}{45,200} - 1 = \underline{\underline{+.11}}$$

FOR DYNAMICS: $V_R = 1286 - 505 = 781$ LB

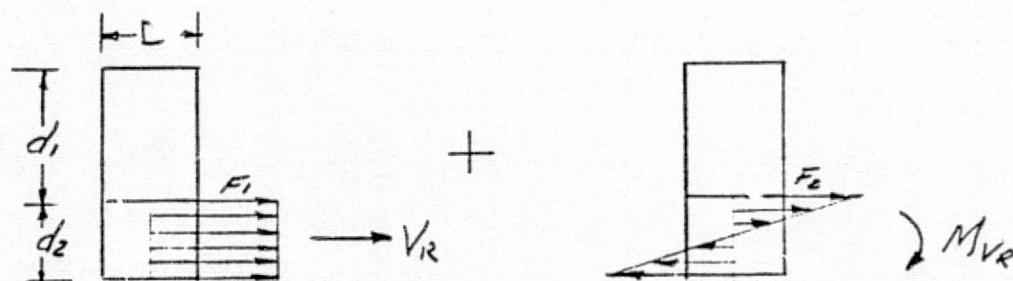
$$\therefore S_{br} = \frac{781}{(.125)(.125)} = 50,000 \text{ psi}$$

FOR $F_{BRY} = 50,000 \text{ psi}$ (NOT BASED ON MICROYIELD)

$$M.S. = \frac{50,000}{50,000} - 1 = \underline{\underline{0}}$$

A MORE CONSERVATIVE APPROACH WOULD BE
TO INCLUDE LOCAL BENDING IN THE ANALYSIS

THUS, WE HAVE THE FOLLOWING LOAD
DISTRIBUTION



$$\text{THUS, } F_{MAX} = F_1 + F_2 = \frac{V_R}{d_2} + \frac{3V_R}{d_2} = \frac{4V_R}{d_2}$$



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BEARING IN ISP FACE SHEETS (CONT.)

$$\text{SINCE } S_{br} = \frac{F}{D}, \text{ THEN } \underline{S_{br} = \frac{4V_R}{d_2 D}}$$

ACCOUNTING FOR FRICTION EFFECTS WITH $\mu = .25$, THE
REQUIRED CLAMP LOAD NECESSARY TO MAINTAIN
A ZERO MARGIN OF SAFETY IS 4370 LB.

$$\therefore F_f = .25(4370) = 1092. \text{ LB}$$

$$\text{FOR STATIC LOADING; } V_R = 1211 - 1092 = 119. \text{ LB}$$

$$\therefore S_{br} = \frac{4(119)}{(.125)(.125)} = 30,500 \text{ psi}$$

FOR $F_{BRY} = 50,000 \text{ psi}$ (NOT BASED ON MICROFIELD)

$$M.S. = \frac{50,000}{30,500} - 1 = \underline{\underline{+.64}}$$

$$\text{FOR DYNAMIC LOADING, } V_R = 1286 - 1092 = 194. \text{ LB}$$

$$\therefore S_{br} = \frac{4(194)}{(.125)(.125)} = 49,670 \text{ psi}$$

$$\text{FOR } F_{BRY} = 50,000 \text{ psi}$$

$$M.S. = \frac{50,000}{49,670} - 1 = \underline{\underline{+0.}}$$

NEED CLAMP LOAD
OF 4370. LB
($\mu = .25$)

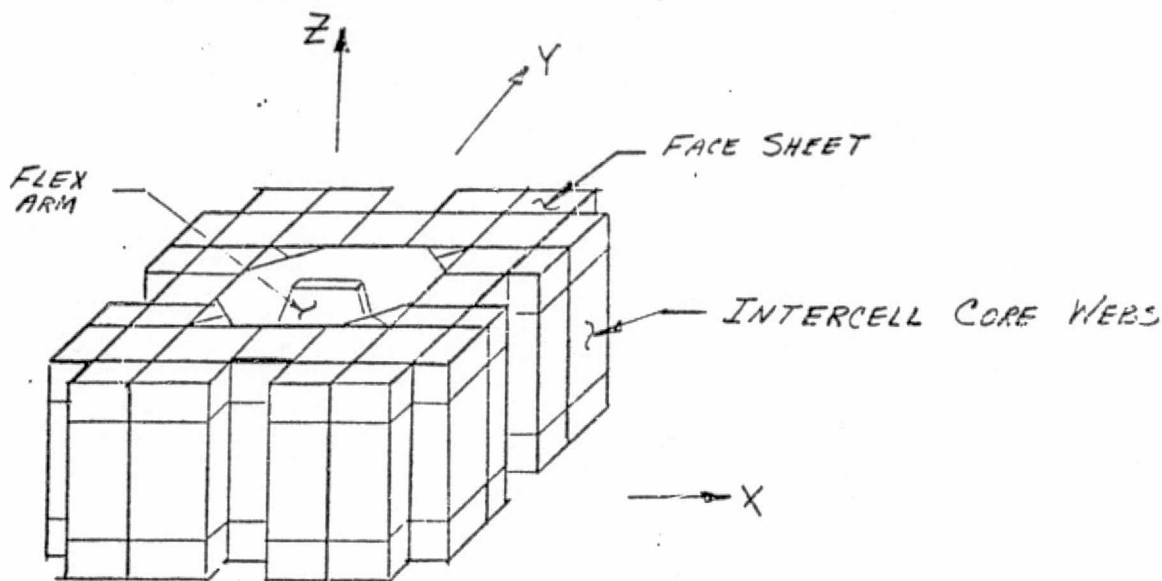


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Approved: _____, _____ Project: SMM
Date: _____ Contract No. _____
Subject: _____ Page No. 42 of _____

15-1 BENDING IN ISP FACE SHEETS



FROM THE DETAILED FINITE ELEMENT MODEL OF THE
FLEX MOUNT, PLATE STRESSES ARE OBTAINED

FOR STATIC LOADING; $17.8g \text{ X} + 3.3g \text{ Y}$

$$S_b = 13,400 \text{ psi}$$

FOR $F_{by} = 18,000 \text{ psi}$ (6061 T6 MICROYIELD)

$$M.S. = \frac{18,000}{13,400} - 1 = \underline{\underline{+.34}}$$

FOR DYNAMIC LOADING: $3.84g \text{ X}$ PANEL BASE INPUT @ 39. Hz

$$S_b = 14,400 \text{ psi}$$



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BENDING IN ISP FACE SHEETS (CONT.)

AGAIN, For $F_{BY} = 18,000 \text{ psi}$ (MICROYIELD)

$$M.S. = \frac{18,000}{14,400} - 1 = \underline{\underline{+.25}}$$



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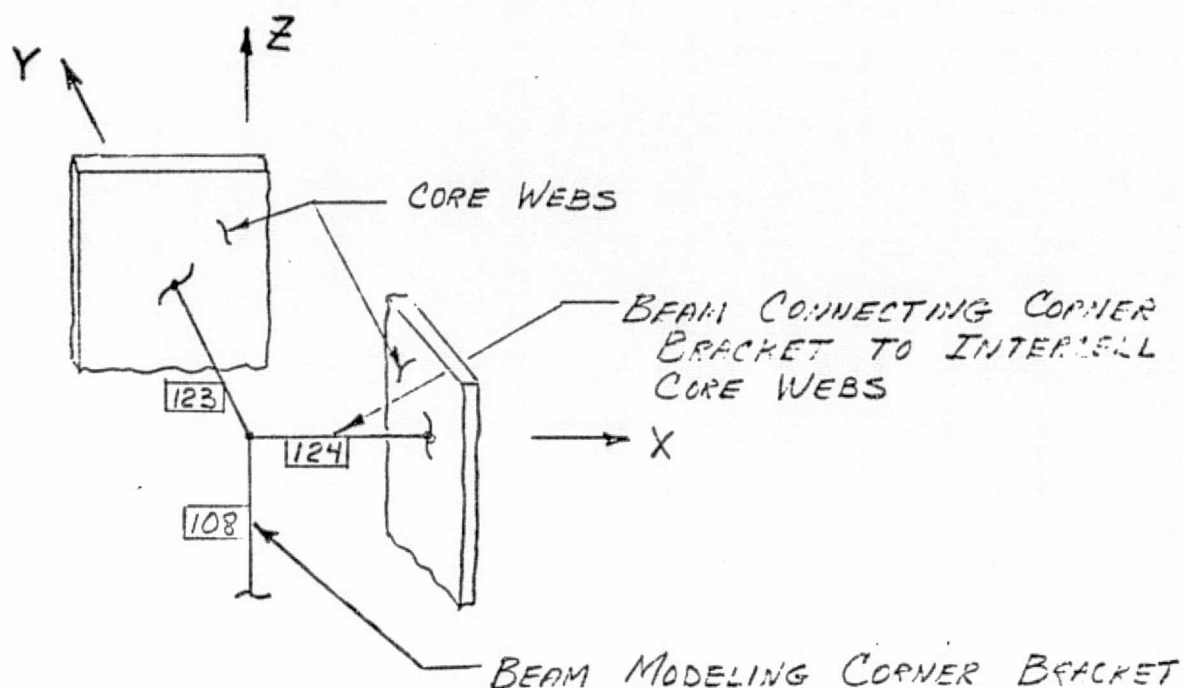
Prepared for: _____ Report No. 16-1.1

Approved: _____, _____ Project: SMM

Date: _____ Contract No. _____

Subject: _____ Page No. 45 of _____

16-1 CORNER BRACKET SCREW LOADS - TENSION



FOR STATIC LOADING: $17.8g X + 3.3g Y$

$$P_x = 49. LB$$

FOR TENSILE LOAD CAPACITY = $2000 LB^*$ (#10-32 Titanium)

$$\therefore M.S. = \frac{2000}{49} - 1 = \underline{\text{AMPLE}}$$

FOR DYNAMIC LOADING: $3.84g X$ PANEL BASE INPUT @ 39 Hz

$$P_x = 51. LB$$

$$\therefore M.S. = \frac{2000}{51} - 1 = \underline{\text{AMPLE}}$$

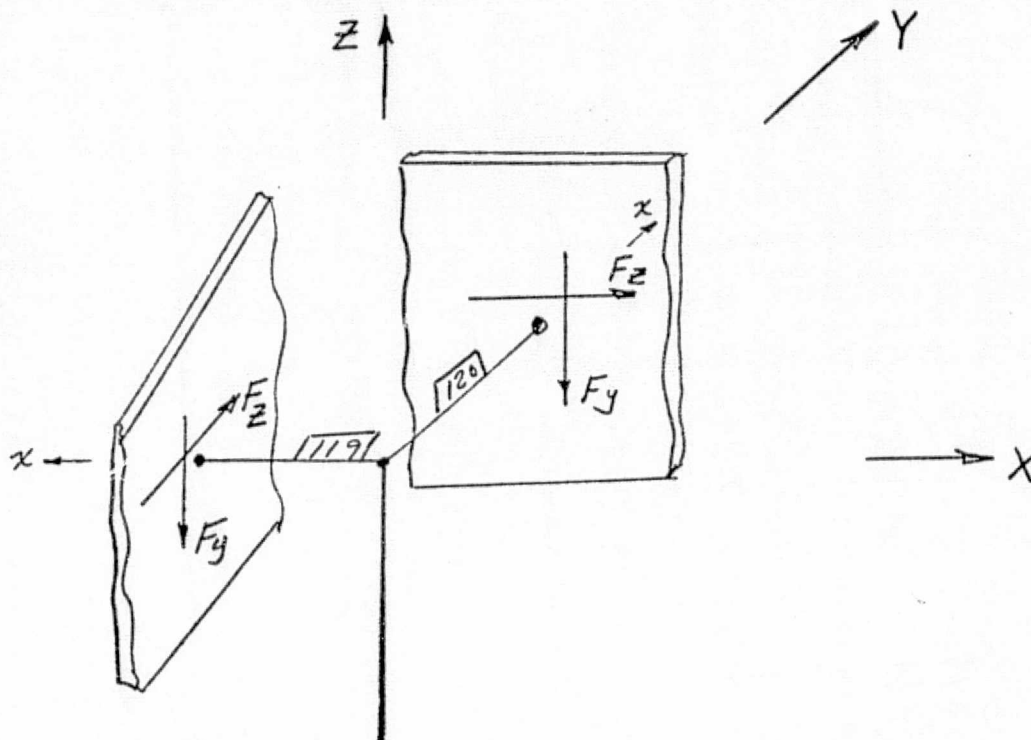
* CLAMP LOAD REQUIRED TO PREVENT SLIP (SECT. 17-1)



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Subject: _____ Page No 4/6 of _____

17-1 CORNER BRACKET-REQUIRED FRICTION LOADS



FOR STATIC LOADING: $17.8g X + 3.3g Y$

$$(F_y)_{119} = 331. \text{ LB} \quad (F_z)_{119} = -149. \text{ LB}$$

(NOTE: LOADS IN BEAM 120 ARE LESS)

IN GLOBAL COORDINATES, $F_y = -149$, $F_z = -331$

$$\therefore V_R = [(149)^2 + (331)^2] = 363. \text{ LB}$$

FOR TENSILE LOAD CAPACITY = 2000 LB (#10-32 Titanium), $F_t = .2(2000) = 400 \text{ LB}$

$$\therefore M.S. = \frac{400}{363} - 1 = \underline{\underline{+1.10}}$$



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CORNER BRACKET, REQUIRED FRICTION LOADS (CONT.)

FOR DYNAMIC LOADING: $3.84g \times \text{PANEL BASE INPUT @ } 39. \text{ Hz}$

$$V_R = 386. \text{ LB}$$

AGAIN FOR TENSILE LOAD CAPACITY = 2000. LB

$$F_f = .2(2000) = 400 \text{ LB.}$$

$$\therefore M.S. = \frac{400}{386} - 1 = \underline{\underline{+0.03}}$$

APPENDIX F

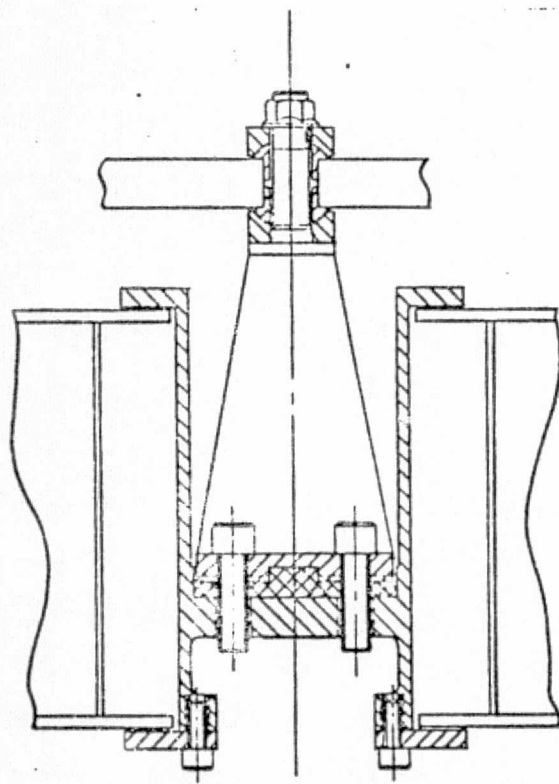
MID-POINT REVIEW VUGRAPHS

- DETAIL STUDY OF ERROR BUDGET REQUIREMENTS.
- REVIEW NASA MECHANICAL ANALYSIS - 'MEGA' ANALYSIS.
- REVIEW NASA THERMAL ANALYSIS AND EXPECTED EFFECTS ON ALIGNMENT.
- DEVELOP METHODS OF COALIGNMENT AND REQUIRED MOUNTING HARDWARE ACCURACIES.
- DEVELOP OPTICAL CUBE AND RELATIVE MIRROR POSITIONS.
- REVIEW NASA DETAIL DESIGN OF FLEX AND FIXED MOUNTS.
 - THERMAL ANALYSIS
 - MECHANICAL ANALYSIS
 - FABRICATION TECHNIQUES AND TOLERANCES
- REVIEW OAO AND ATM EXPERIMENT MOUNTS
 - DETAIL DESIGNS
 - METHODS OF ALIGNMENT
 - EXPECTED ACCURACY AND FLIGHT HISTORY
- DESIGN SUITABLE EXPERIMENT MOUNTS.
- DEVELOP ENVIRONMENTAL TEST PROGRAM.
- DETAIL STUDY OF NASA ETM DESIGN.
- DETAIL MECHANICAL ANALYSIS OF AN INDIVIDUAL ETM PLATFORM CELL AND FLEX MOUNT ASSEMBLY.

- SPHERICAL SEAT
 - ALIGNMENT BY TEMPLATE
 - ALIGNMENT BY ADJUSTMENT
- PLANAR SEAT
- INTERMEDIATE PLATE

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FLEX MOUNT (PRESENT DESIGN)



RECOMMENDED CHANGES

- BLADE THICKNESS
- CROSSED KEYS
- HOUSING BORE
- FLEX MEMBER SCREWS
- BALL SEAT/FLEX MEMBER -
MATCHED SET
- WITNESS MARK KEYING
- BOTTOM PLATE/HOUSING -
DOWELS
- BALL AND SEAT DIMENSIONS
AND TOLERANCES

REASONS

- STRESS
- PRODUCIBILITY
- PRODUCIBILITY
- STRESS
- ALIGNMENT ERROR BUDGET
- ALIGNMENT ERROR BUDGET
- ALIGNMENT MAINTENANCE
- ALIGNMENT ERROR BUDGET



MOUNTING ERROR SOURCES (PRESENT SPHERICAL SEAT)

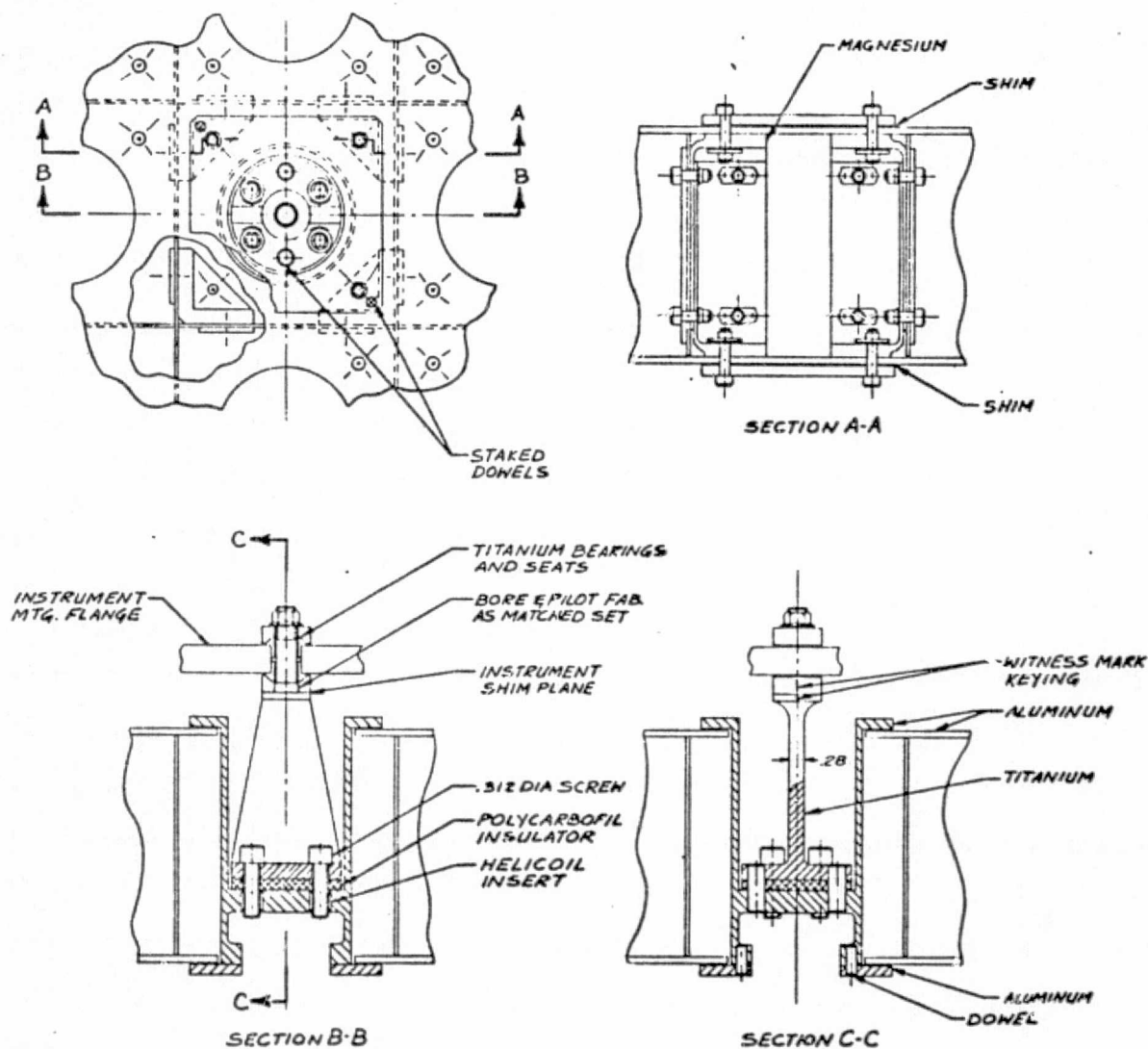
INTERFACE	DIAMETRICAL ALLOWANCE (IN.)	DIAMETRICAL TOLERANCE (IN.)	MAXIMUM DIAMETRICAL CLEARANCE (IN.)	MAXIMUM SHIFT* (ARCSEC)
SPHERICAL SEAT/SEAT BORE	⊙.0002 (ESTIMATE)		.0002 (EQUI- VALENT)	6.9
SEAT BORE/FLEX PILOT	.0005 (ESTIMATE)	.0005 SEAT (ESTIMATE) .0005 FLEX	.0015	51.6
INSULATOR PILOT/HOUSING BORE	.0005 INTER- FERENCE	.0005	.0005	17.2

DOES NOT INCLUDE ERROR BETWEEN INSTRUMENT CUBE AND INSTRUMENT MOUNTING HOLE PATTERN
WHICH IS INCLUDED IN MAXIMUM ALLOWABLE MOUNTING ERROR OF ± 10 ARCSEC.

*BASED ON 12 INCH SPACING BETWEEN MOUNTS.

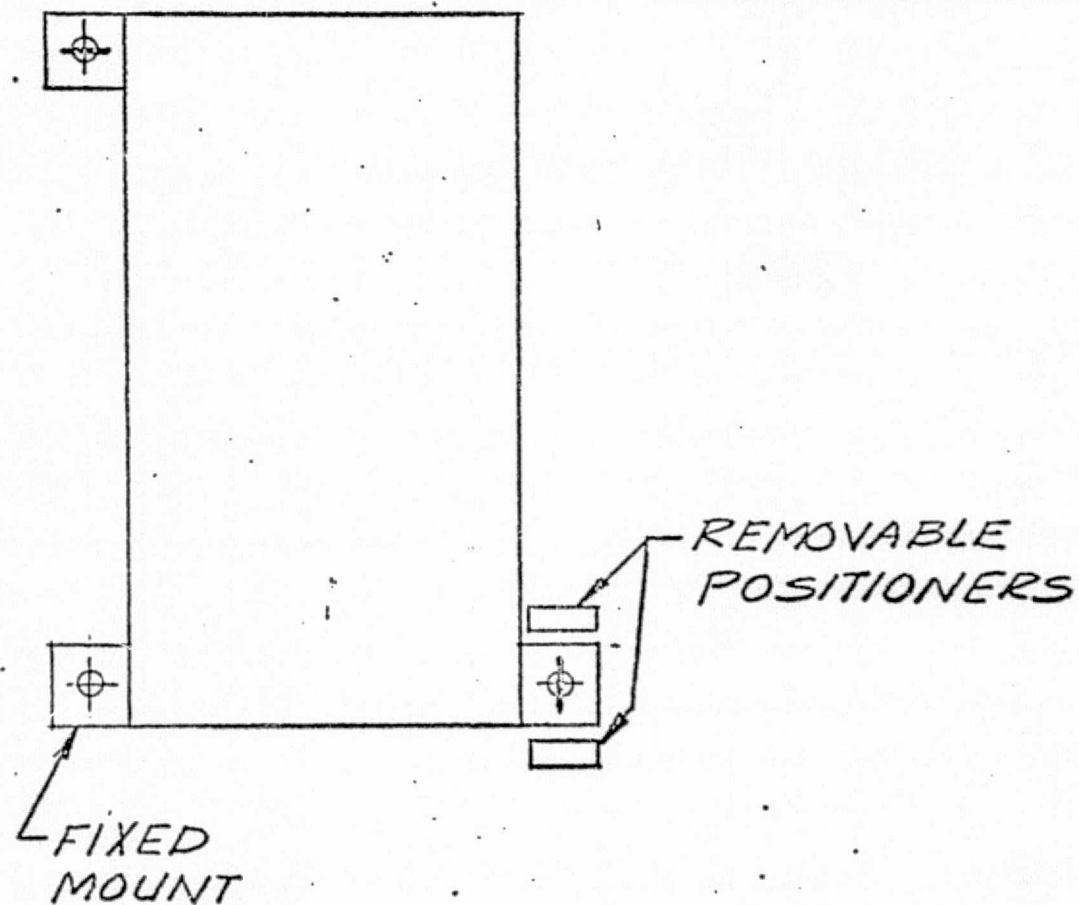
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FLEX MOUNT ASSEMBLY (SPHERICAL SEAT)



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OPTICAL ALIGNMENT ADJUSTMENTS



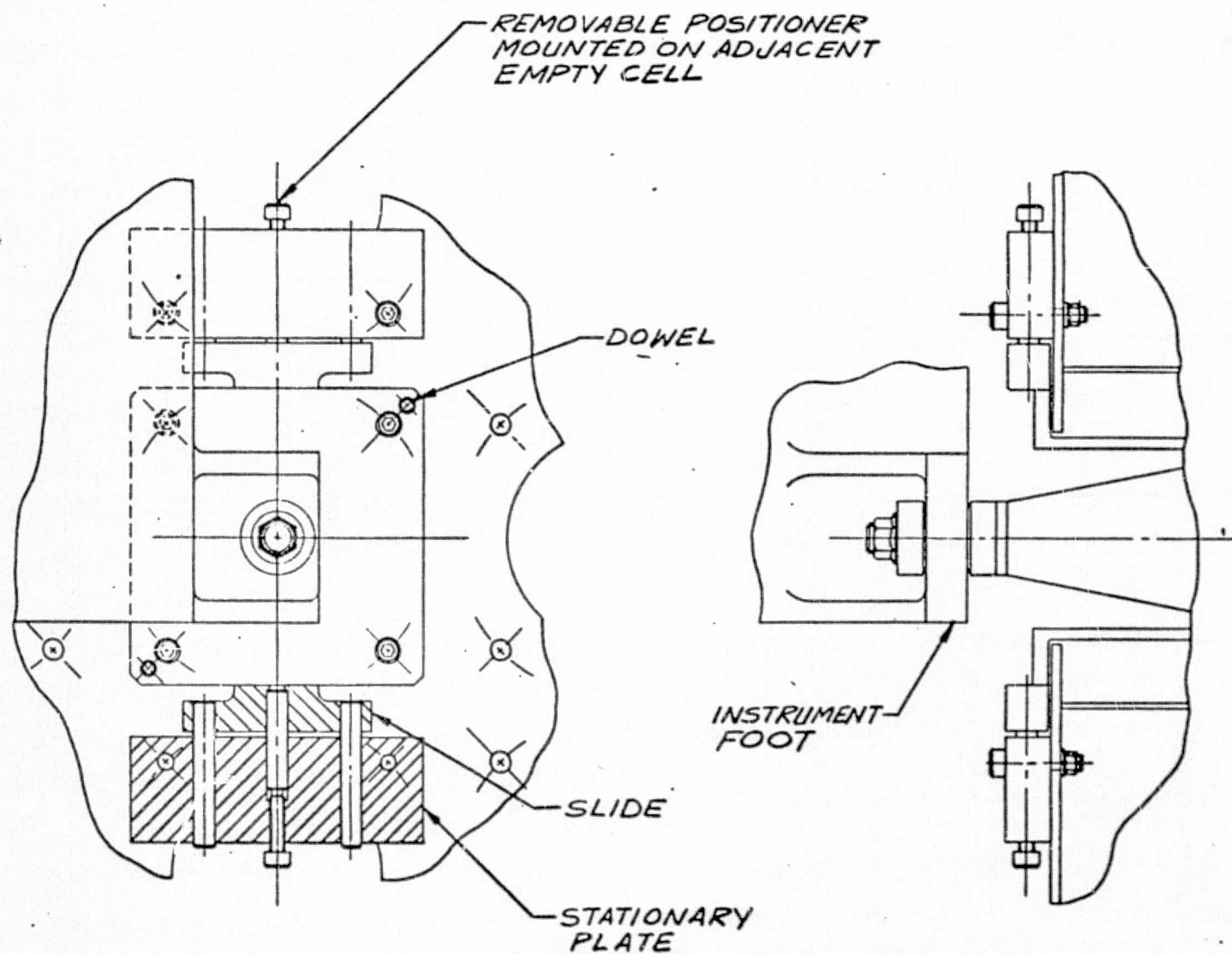


ALIGNABLE SPHERICAL MOUNT

- BALL SEAT BORE MATCHED TO PILOT ON STUB
0 - 50 μ IN DIAMETRAL CLEARANCE
- ALIGNMENT BY ROTATING ABOUT FIXED MOUNT
- FLEX MOUNTS ADJUSTED BY MOVING ENTIRE PRE-SHIMMED HOUSING IN ISP
- INSTRUMENT SHIMMING AT INTERFACE BELOW BALL SEAT
- HOUSINGS TORQUED TO ISP DURING SYSTEM OPTICAL ALIGNMENT

PROBLEMS

- ACCESS FOR TORQUING DURING OPTICAL ALIGNMENT
- INSTRUMENTS MUST BE REMOVED TO DOWEL HOUSINGS TO ISP





ALIGNABLE PLANAR MOUNT

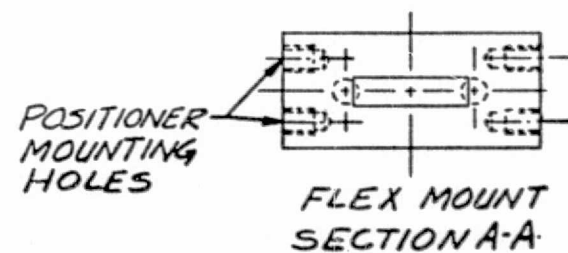
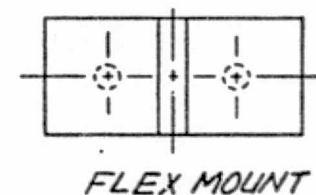
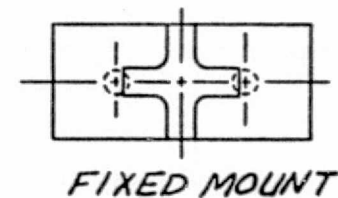
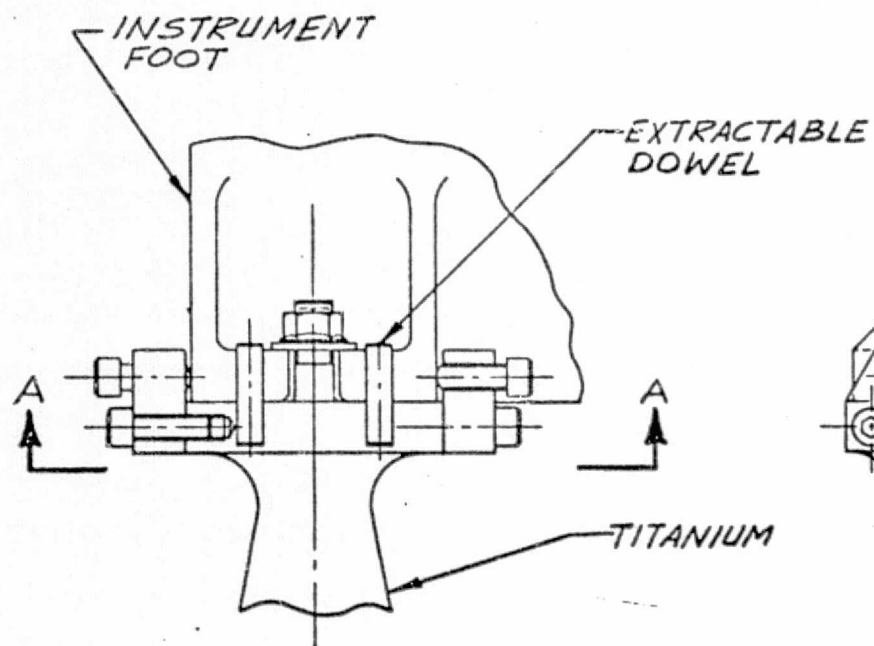
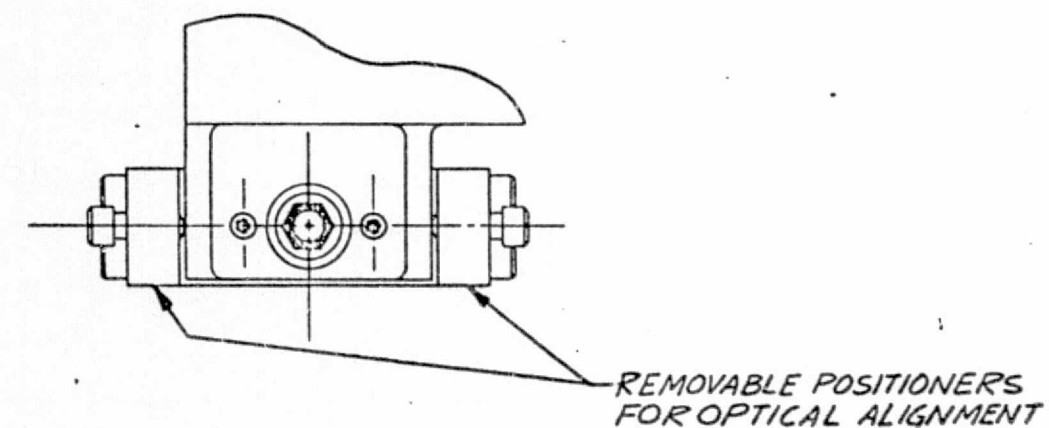
- ALIGNMENT ADJUSTMENTS AT INSTRUMENT FLANGE
- TEMPORARY POSITIONER MOUNTED TO ONE FLEX MOUNT
- INSTRUMENT SHIMMING BELOW INSTRUMENT FLANGE
- INSTRUMENT FLANGE PINNED TO MOUNTS FOLLOWING ALIGNMENT

PROBLEMS

- MATERIAL REMOVAL IN OAF
- DOWEL EXTRACTION

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PLANAR SEAT



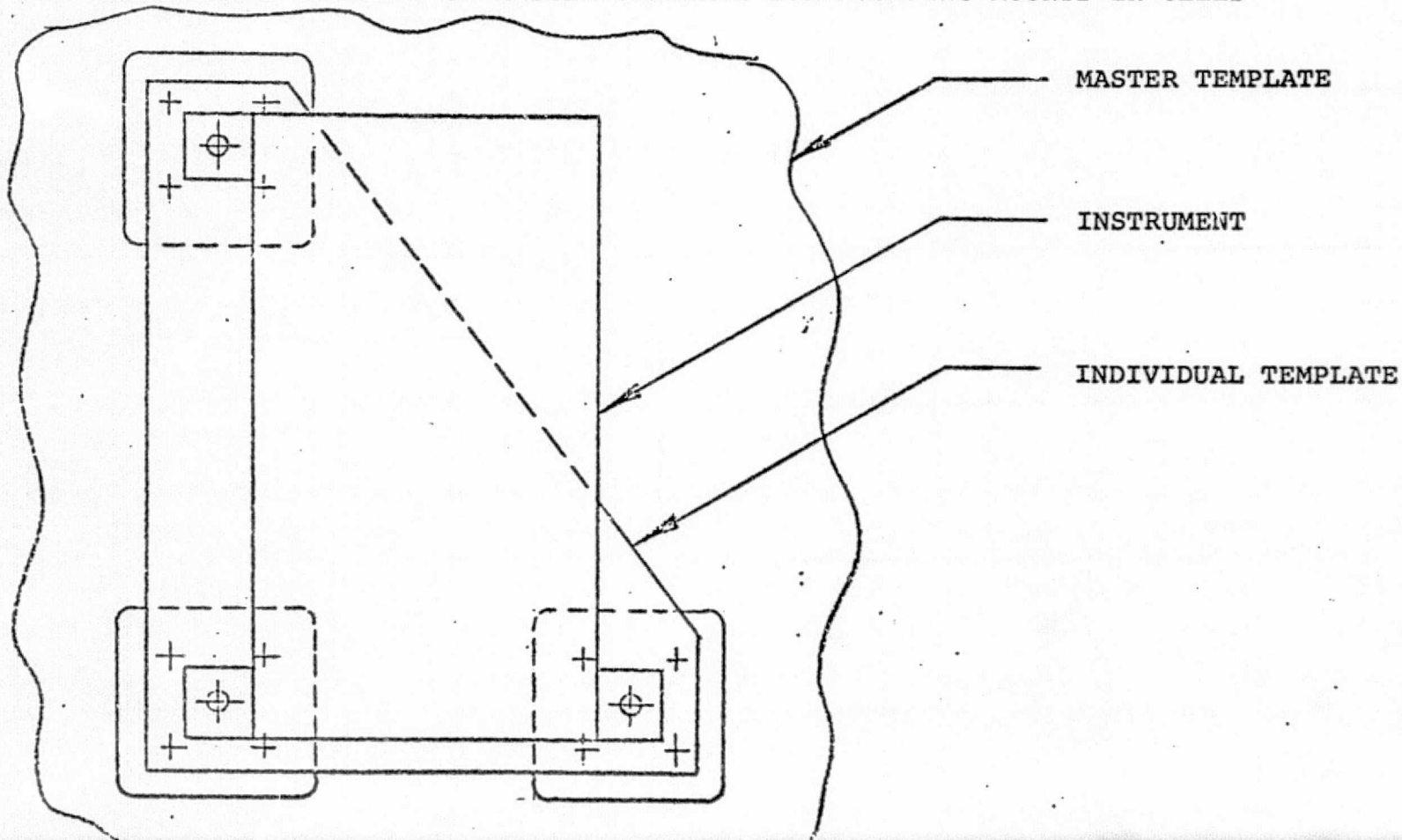
INTERMEDIATE PLATE CONCEPT

- PLATE OR STRUCTURE IS PINNED TO PLANAR FLEX AND FIXED MOUNTS.
- PLATE IS PART OF INSTRUMENT AND IS DRILLED FROM TEMPLATE.
- INSTRUMENT IS ALIGNABLE ON PLATE WITH FEET PINNED TO PLATE FOLLOWING SYSTEM ALIGNMENT.
- DESIGN ALLOWS FOR FLEXIBILITY IN INSTRUMENT LOAD PATH AND MOUNTING FOOT LOCATIONS.
- DESIGN ALLOWS FOR FLEXIBILITY IN CELL SELECTION FOR MOUNTING INTERMEDIATE PLATE TO ISP.
- ALIGNMENT MECHANISMS CAN BE STANDARDIZED, REMOVABLE, AND LOCATED TO SUIT THE REQUIREMENTS OF EACH INSTRUMENT.
- THIS CONCEPT FOR ANY INDIVIDUAL INSTRUMENT IS COMPATIBLE WITH THE OTHER CANDIDATE ALIGNMENT CONCEPTS.



ALIGNMENT BY TEMPLATE

- MASTER TEMPLATE MOUNTED TO ISP (SCREWS AND DOWELS)
- MASTER TEMPLATE MOUNTS INDIVIDUAL TEMPLATES (SCREWS AND DOWELS)
- INDIVIDUAL TEMPLATE FOR DRILLING INSTRUMENT FLANGE
- INDIVIDUAL TEMPLATE ON MASTER TEMPLATE FOR LOCATING MOUNTS IN CELLS



TEMPLATE ALIGNMENT DEFICIENCIES

- SPHERICAL BEARINGS IN INSTRUMENT FLANGE MUST BE INSTALLED WITHIN THE ± 10 ARCSECOND MOUNTING BUDGET.
- ALIGNMENTS INCLUDING FSS WILL BE TO MASTER TEMPLATE RATHER THAN FSS.
- CUMULATIVE TOLERANCES IN REFERENCE TRANSFERS BETWEEN BORESIGHT, INSTRUMENT MIRROR, INDIVIDUAL TEMPLATE AND MASTER TEMPLATE.
- ALLOWANCES AND TOLERANCES IN BOLTED JOINT PIECE PARTS AT THE MOUNTING POINTS.
- GRAVITY EFFECTS MASK RELATIVE ALIGNMENTS AND DO NOT ALLOW FOR ALIGNMENT VERIFICATION OF THE COMPLETED EXPERIMENT PACKAGE USING INSTRUMENT MIRRORS.



ALIGNMENT OPTIONS

- TEMPLATE ONLY
- OPTICAL ALIGNMENT
- GRAVITY COMPENSATION (INVERSION FIXTURE)
- GRAVITY COMPENSATION (LOAD CELLS)

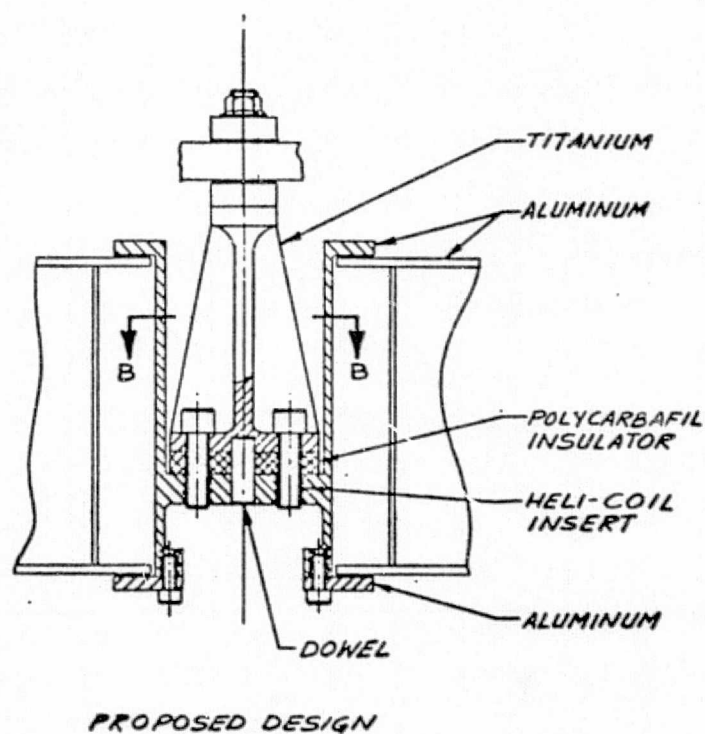
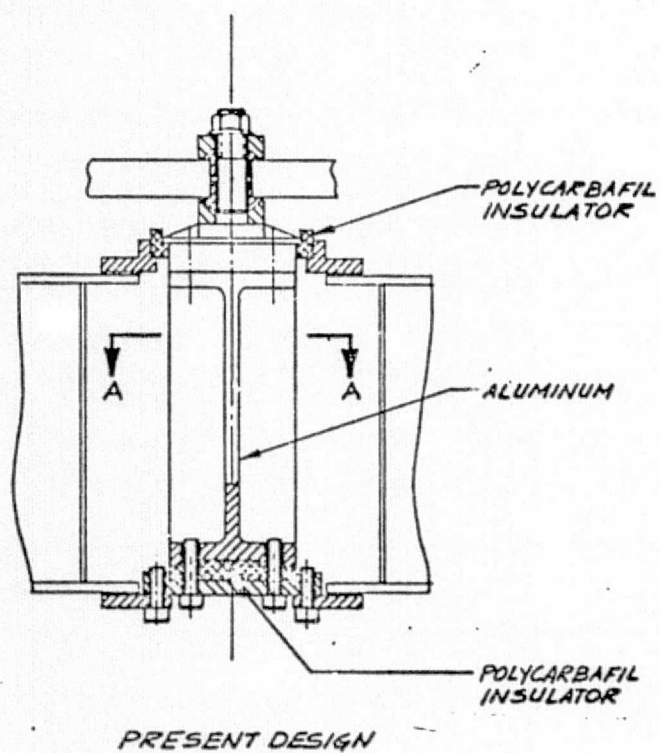
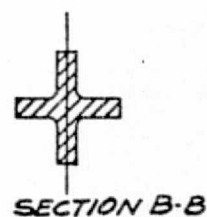
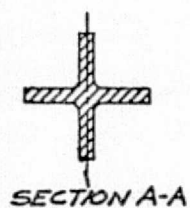
RECOMMENDATIONS

- OPTICAL ALIGNMENT (SYSTEM LEVEL)
- ALL INSTRUMENTS MOUNTED PRIOR TO TORQUING ANY MOUNTING HARDWARE
- INVERSION FIXTURE FOR GRAVITY COMPENSATION
- NO LOAD CELLS
- MICRO-ADJUSTMENT AT INSTRUMENT FOOT

- MOUNT MASTER TEMPLATE TO ISP WITH SCREWS AND DOWELS.
- MOUNT INSTRUMENT TEMPLATES TO MASTER TEMPLATE WITH SCREWS AND DOWELS.
- DRILL MOUNT HOLES IN ALL INDIVIDUAL TEMPLATES.
- DRILL INSTRUMENT FLANGES FROM INSTRUMENT TEMPLATES.
- SHIM AND INSTALL MOUNTS TO ISP USING INSTRUMENT AND MASTER TEMPLATES.
- MOUNT ISP IN OAF INVERSION FIXTURE IN UPRIGHT ORIENTATION.
- MOUNT ALL INSTRUMENTS TO ISP.
- MEASURE ALIGNMENTS WITH ISP UPRIGHT.
- MEASURE ALIGNMENTS WITH ISP INVERTED.
- CALCULATE SHIM SIZES AND THEODOLITE SETTINGS FOR UPRIGHT ORIENTATION
BASED ON ZERO G AVERAGES.
- INSTALL SHIMS AND POSITIONERS AND ALIGN EACH INSTRUMENT TO CALCULATED
SETTINGS WITH INVERSION FIXTURE UPRIGHT.
- MEASURE ALIGNMENTS WITH ISP INVERTED AND ITERATE ALIGNMENT SEQUENCE.
- REMOVE ALL INSTRUMENTS FROM ISP AND DOWEL ALL MOUNT HOUSINGS AND BOTTOM PLATES.
- MOUNT ALL INSTRUMENTS TO ISP AND VERIFY ALIGNMENTS WITH ISP UPRIGHT.

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FIXED MOUNT



- REVISE FIXED MOUNT CONFIGURATION
- ELIMINATE CROSSED KEYS IN POLYCARBAFIL AND HOUSING
- FABRICATE STUB PILOT AND BALL SEAT BORE AS MATCHED SET
- INCREASE SIZE OF INTERNAL MOUNTING SCREWS



SMM/EXPERIMENT INTERFACE SPECIFICATION

4.2.2 MOUNTING HOLE POSITION

"....THE INSTRUMENTOR SHALL DETERMINE THE OPTICAL AXIS OF THE INSTRUMENT AND THE POSITION OF THE OPTICAL REFERENCE SURFACES RELATIVE TO THE MOUNTING HOLE PATTERN WITH TBD PRECISION."

4.3.1 ALIGNMENT RELATIONSHIP

"....THE ANGLE BETWEEN THE INSTRUMENT OPTICAL AXIS AND THE NORMAL TO THE PRIME OPTICAL REFERENCE SURFACE SHALL NOT EXCEED 3 ARC MINUTES...."

4.3.2 ALIGNMENT MEASUREMENT

"THE DEVICES ON THE INSTRUMENT SUPPORT PLATE WILL BE PRE-ADJUSTED TO PRODUCE CO-ALIGNMENT OF ALL INSTRUMENTS TO THE SPACECRAFT REFERENCE TO ± 10 ARCSEC....NO ATTEMPT WILL BE MADE TO MEASURE OR COALIGN ACTUAL INSTRUMENT BORESIGHTS OR TO MEASURE BY SOURCE STIMULATION...."



ISP THERMAL OVERVIEW

INSTRUMENT SUPPORT PLATE

- CALCULATED STEADY STATE DISTORTIONS OF PLATE FROM EXTERNAL ENVIRONMENT IS IDENTICAL WITH THAT STATED BY NASA.
- OVERALL PLATE DISTORTION DUE TO CONDUCTANCE THRU MOUNTS IS NEGLIGIBLE EVEN WHEN POLYCARBAFIL INSULATION IS DELETED.
- APPROX. CALCULATIONS INDICATE IT WILL BE DIFFICULT TO MAINTAIN PLATE WARP TO ± 1 SEC IN 5 MINUTES.

INSTRUMENT MOUNTINGS

- INSTRUMENT PITCH AND ROLL ANGLE ERRORS ARE VERY SENSITIVE TO THE DIFFERENTIAL TEMPERATURE BETWEEN THAT OF ALIGNMENT AND ORBIT OPERATION UNLESS SUPPORT MEMBER MATERIALS AND PATH LENGTHS ARE SIMILAR.
- EVEN WITH IDENTICAL MOUNTS, THERMAL GRADIENTS IN THE INSTRUMENT PROPER CAUSE LARGE ERRORS IN PITCH AND ROLL AND EITHER STRESS OR ERROR IN THE YAW DIRECTION.
- GRADIENTS IN THE MOUNTS CAUSE MOST OF THE PROBLEMS WHICH COULD BE ELIMINATED IF INSTRUMENTS COULD BE INSULATED FROM THE MOUNT.

ISP: DISTORTION, Q AND ΔT RELATIONSHIPS

ISP FACE-FACE CONDUCTANCE:

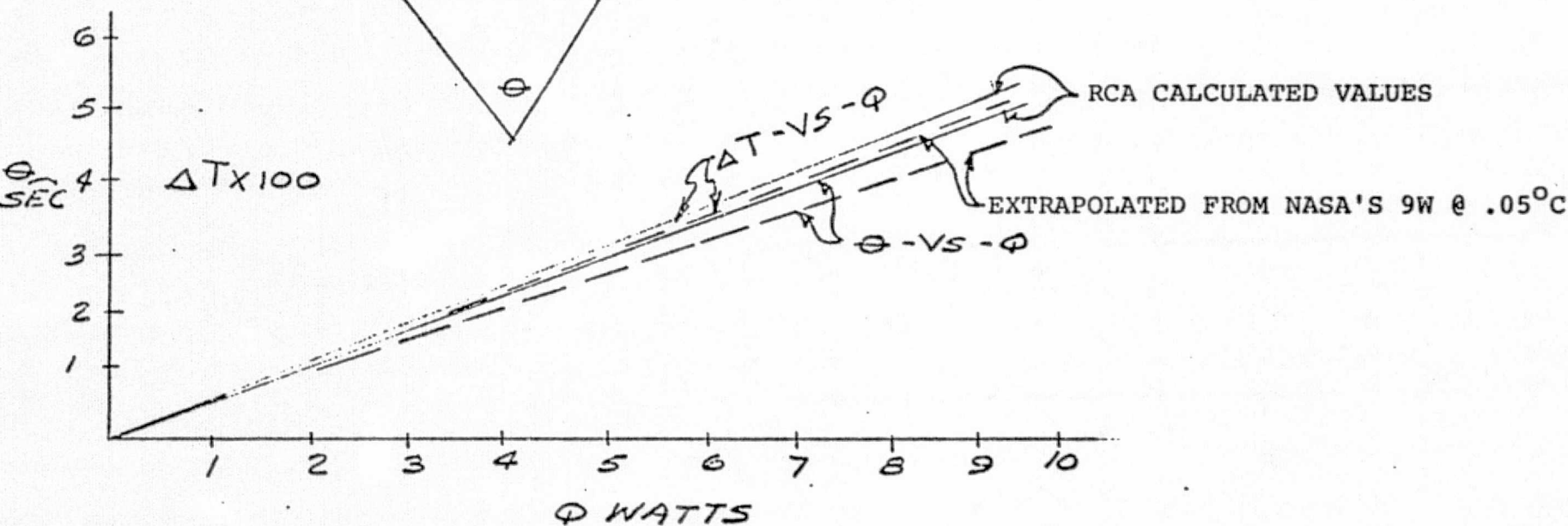
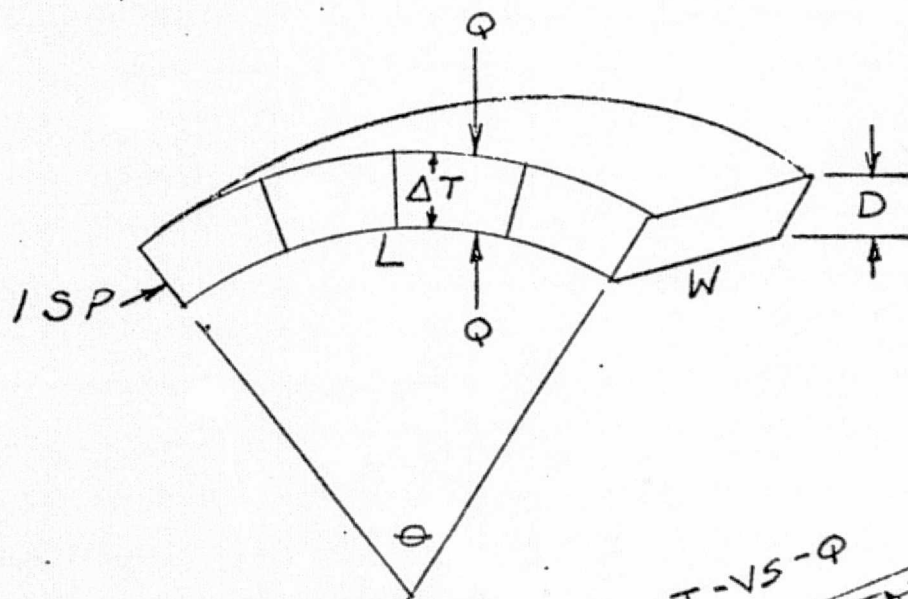
$$K_{IJ} = .137 \frac{W}{^{\circ}C}$$

$$\text{NEWTONIAN RAD. COUPLING } K_{EQ} = .012 \frac{W}{^{\circ}C}$$

$$Q = \frac{W\theta}{\alpha} (K_{IJ} + K_{EQ.})$$

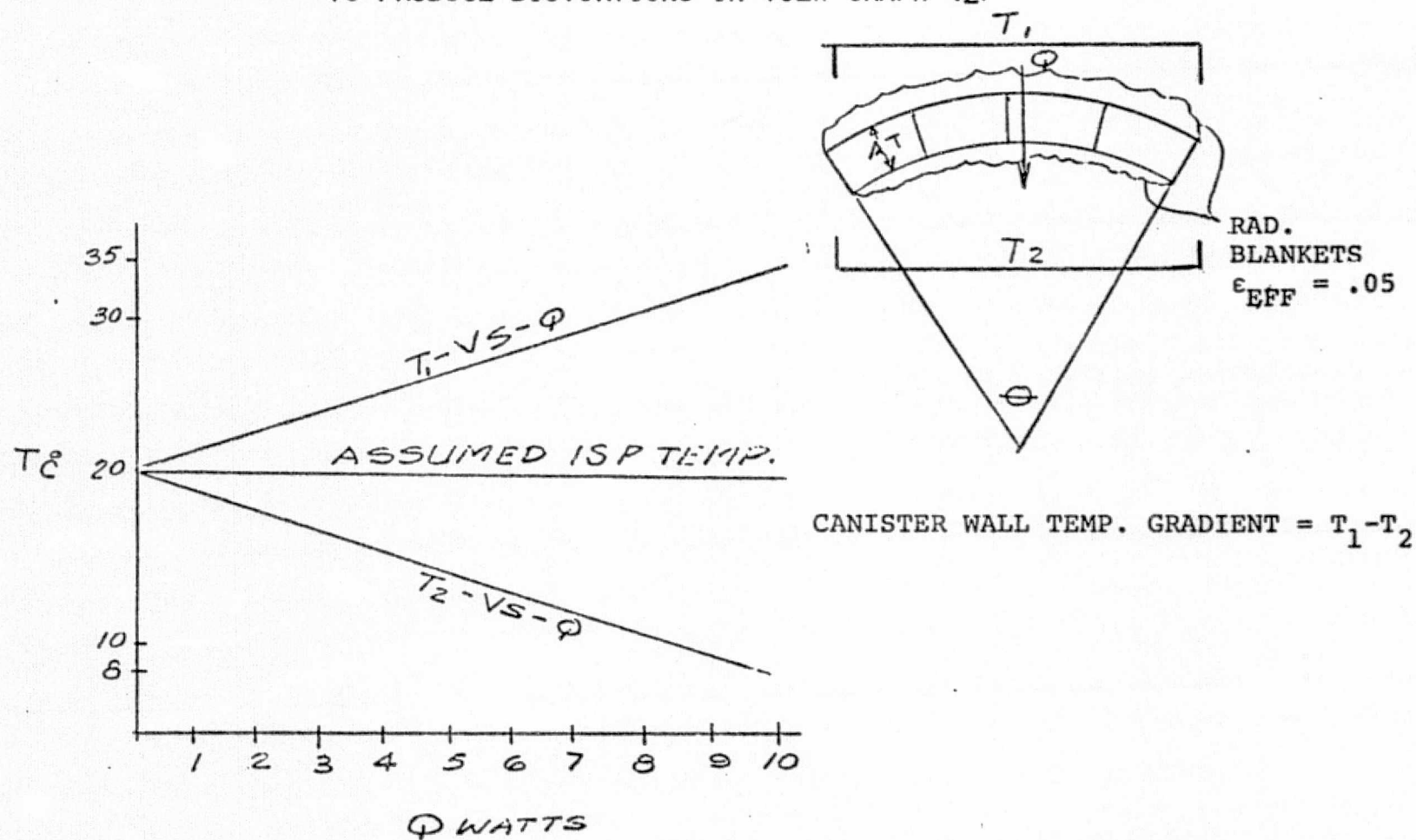
$$\theta = \frac{\alpha Q}{W} \left(\frac{1}{K_{IJ} + K_{EQ.}} \right)$$

$$\Delta T = \frac{DQ}{LW} \left(\frac{1}{K_{IJ} + K_{EQ.}} \right)$$

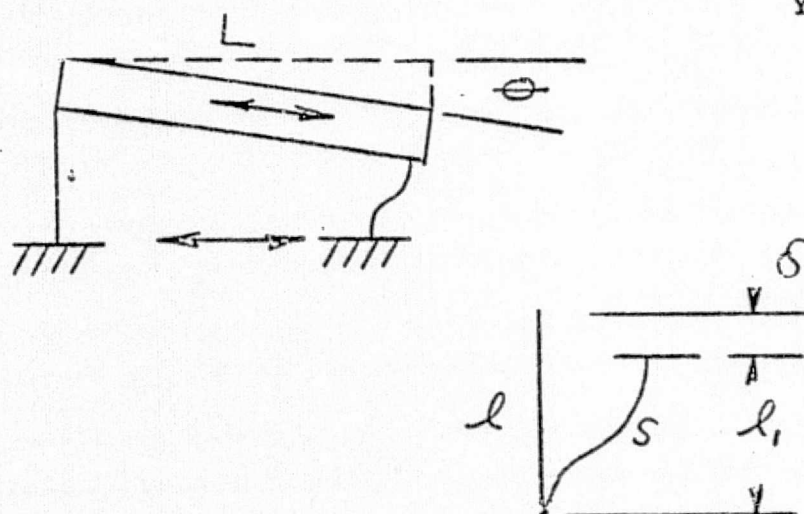


ALLOWABLE CANISTER WALL TEMPERATURE GRADIENTS
TO PRODUCE DISTORTIONS IN VIEW GRAPH (2)

5



PITCH AND ROLL ANGULAR DEFLECTION OF EXPERIMENTS DUE TO DIFFERENTIAL LENGTH GROWTH
OF EXPERIMENTS/ISP



$$y(x) = \frac{L\alpha\Delta T}{3} (\ell - x)^2 (\ell + 2x)$$

(CANTILEVER BEAM GUIDED
ONE END:

STEEL CONST. HANDBOOK
AM. INST. OF STEEL CONS.
5TH ED. 1957, PP 373 EQ. 23)

$$\frac{dy(x)}{dx} = 6 Kx(x - \ell)$$

$$S = \int_0^{\ell} \sqrt{1 + (y')^2} dx$$

IN ORBIT ALIGNMENT ERRORS, SEC

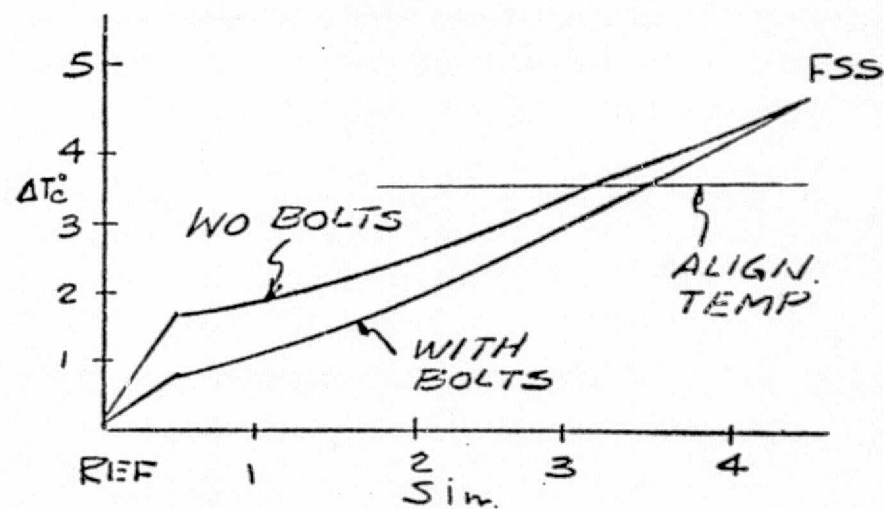
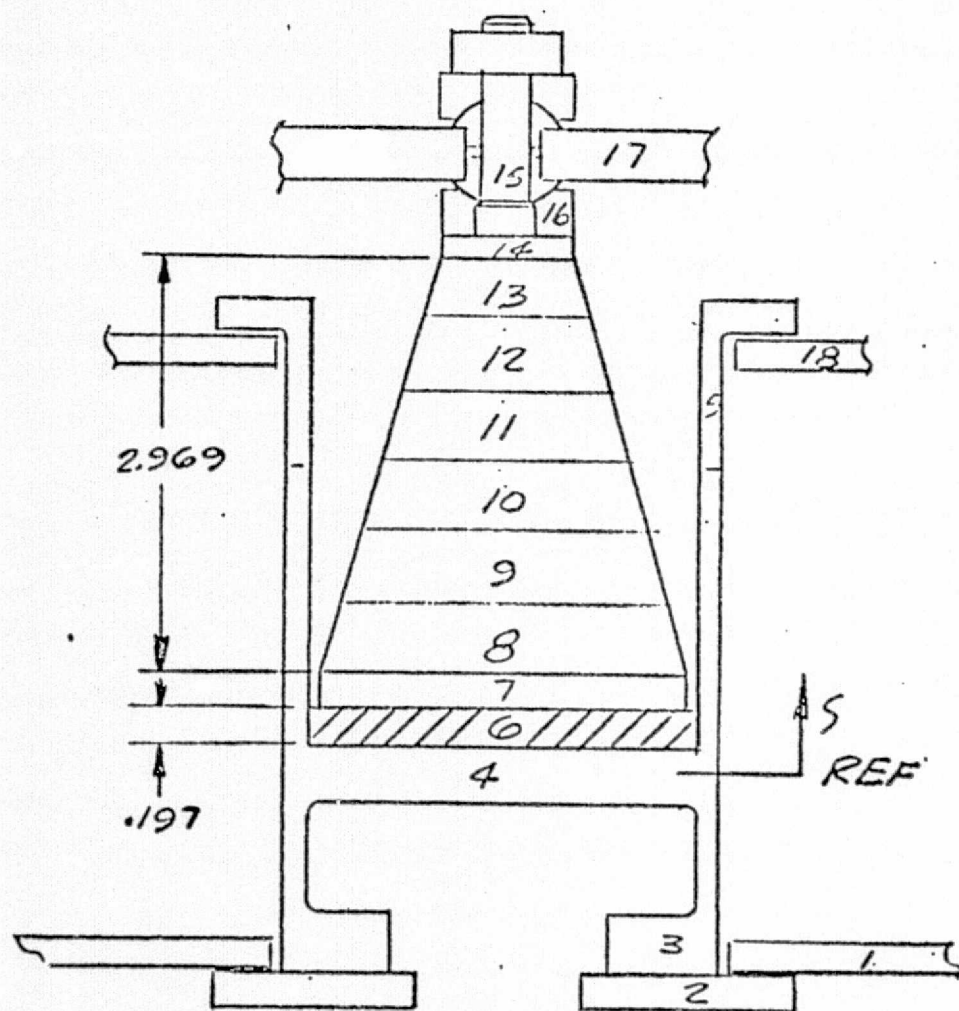
	θ_{PITCH}	θ_{ROLL}
FSS	.003	.002
EX 1	.002	.001
16	.0006	.0002
29	~ 0	.00007
25	.0001	.0001
19	.001	.004

CALCULATE S/ℓ , DETERMINE δ

$$\text{THEN } \theta = \frac{\delta}{L} \frac{360}{2\pi} (3600) \text{ SEC}$$

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THERMAL MODEL OF NASA FLEX MOUNT WITH RESULTANT ORBITAL TEMPERATURE DISTRIBUTION OF FSS MOUNT

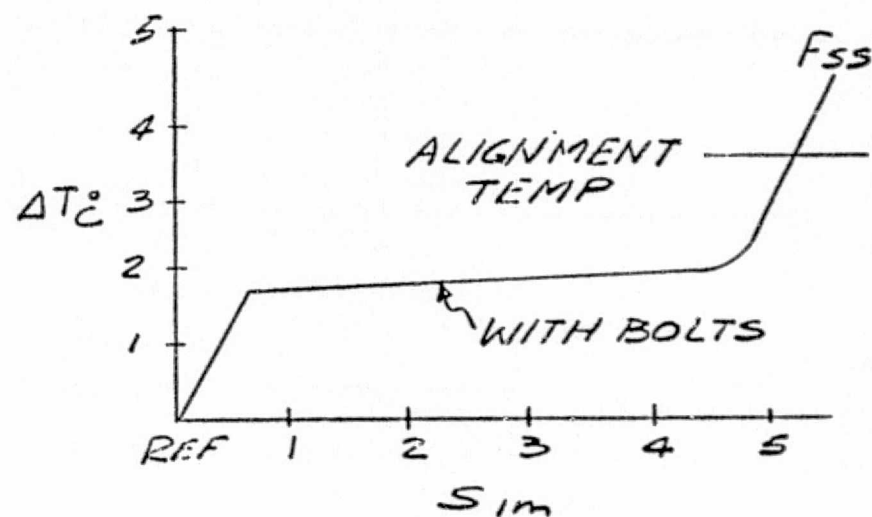
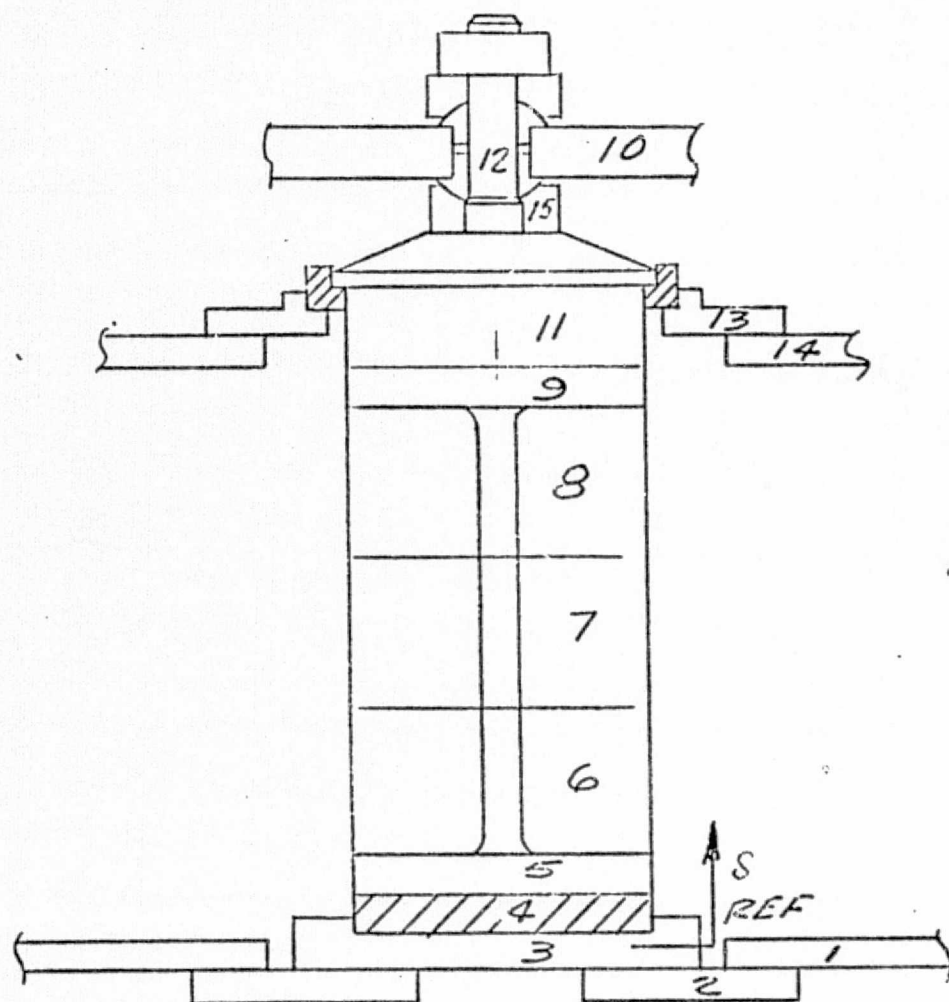


$$K_{\text{eff}} = .017 \frac{\text{W}}{\text{o C}}$$

17-1,18

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THERMAL MODEL OF NASA FIXED MOUNT WITH RESULTANT ORBITAL TEMPERATURE DISTRIBUTION
OF FSS MOUNT

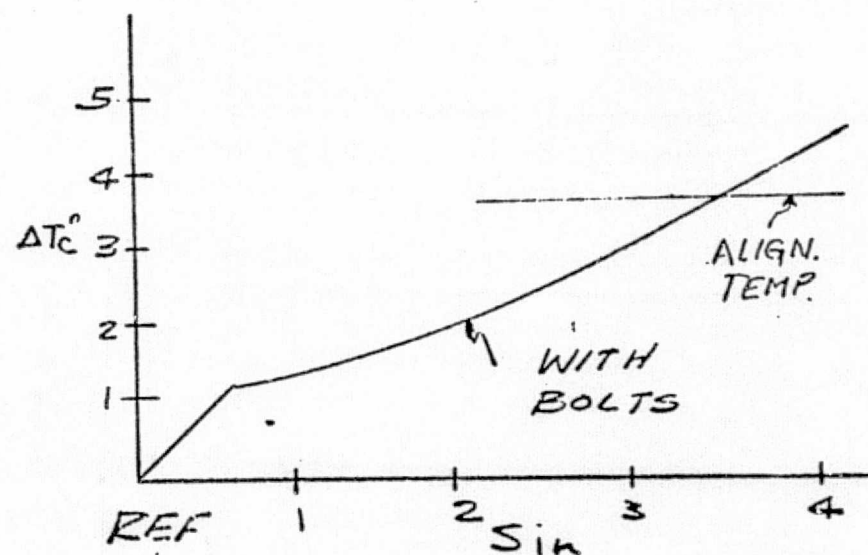
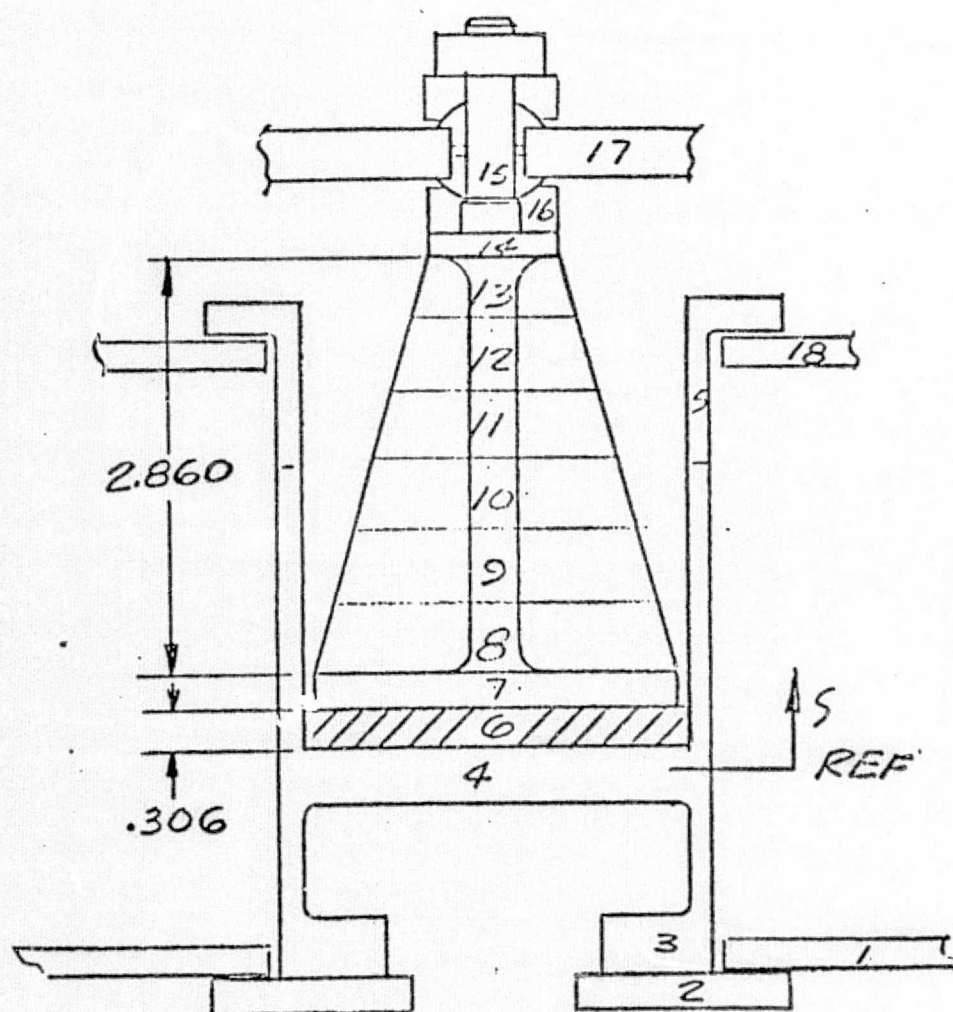


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THERMAL MODEL OF CRUCIFORM FIXED MOUNT WITH RESULTANT ORBITAL TEMPERATURE DISTRIBUTION OF FSS MOUNT

7



$$K_{\text{eff}} 17-1,18 = .017 \text{ w/o } c$$

ORBITAL ALIGNMENT ERRORS AS A RESULT OF FIX-FLEX MOUNT THERMAL GROWTH

$$\theta = \frac{K}{L} \sum_i^j h_i \rho_i \Delta T$$

NASA FIX MOUNT DESIGN, SEC

<u>EXPERIMENT</u>	<u>θ_{PITCH}</u>	<u>θ_{ROLL}</u>	<u>θ_{YAW}</u>
25	-1.49	-1.81	-
29	-1.30	-2.91	-
FSS	-0.61	-2.99	~27 (UNRESTRAINED)

CRUCIFORM FIXED MOUNT DESIGN

25	+0.36	+0.04	-
29	-0.125	-0.172	-
FSS	+0.3	-1.62*	~27 (UNRESTRAINED)

THE RECOMMENDED TITANIUM CRUCIFORM FIXED-MOUNT DESIGN, BECAUSE OF
SIMILARITY TO THE FLEX-MOUNT THERMAL CONFIGURATION, IN THE MAIN, REDUCE
ORBITAL MISALIGNMENTS ABOUT ONE ORDER OF MAGNITUDE OVER THE NASA DESIGN,
HOWEVER, IN SPECIAL CASES (*FSS) THE GAINS ATTRIBUTED TO IMPROVED MOUNT
CHARACTERISTICS ARE OVERWHELMED BY LOCALIZED THERMAL GRADIENTS IN THE
EXPERIMENT PROPER WHICH PRODUCE TEMPERATURE DIFFERENCES IN MOUNTINGS
AND RESULT IN LARGE ALIGNMENT ERRORS.

1. MOUNT ISP IN A VERTICAL PLANE ON THE GRAVITY INVERSION FRAME, DWG. NO. TBD, WHEN THE LATTER IS IN AN UPRIGHT POSITION. THIS FRAME HAS PREVIOUSLY BEEN ATTACHED AND ALIGNED TO THE OAF ROTARY TABLE.
2. MICROPOSITIONERS, DWG. NO. TDB, ARE INSTALLED IN ISP CELLS ADJACENT TO EACH INSTRUMENT'S LOWER FLEX MOUNT. (THE HOUSINGS HAVE BEEN INSTALLED, SHIMMED, ADJUSTED, AND FLANGE BOLTS TIGHTENED USING ALIGNMENT TEMPLATES, DWG. NOS. TBD.)
3. WITH HANDLING FIXTURES, DWG. NOS. TBD, MOUNT ALL EXPERIMENTS AND FSS TO TEMPLATE PREALIGNED AND TIGHTENED, BUT NOT DOWELED MOUNTS. THIS PLACES THE ISP APPROXIMATELY IN THE X-Z PLANE OF A CARTESIAN COORDINATE SYSTEM WITH THE EXPERIMENT'S SUN FACING VECTORS IN THE +X DIRECTION (UPWARD) AND WITH LOADS DISTRIBUTED ON FIXED AND FLEX MOUNTS.
4. MEASURE ALL EXPERIMENTS, ISP, AND FSS, FOR NON-ALIGNMENT BETWEEN THEIR RESPECTIVE OPTICAL CUBE FACE NORMALS, DIRECTLY FOR +Y FACES, AND INDIRECTLY FOR THE +X FACES.

DIRECT +Y CUBE FACE NORMALS ARE RECORDED BY THEODOLITE MEASUREMENT OF: AZIMUTH, ELEVATION, TABLE ROLL, AND DIHEDRAL MIRROR AUTOCOLLIMATION AZIMUTH ANGLES FOR EACH CUBE FACE. THE INDIRECT +X FACE NORMALS ARE RECORDED BY THEODOLITE MEASUREMENT OF: THE AUTOCOLLIMATION ANGLES OF APPROPRIATE UPPER RELAY MIRROR(S), THEIR AZIMUTH AND ELEVATION ANGLES, THE AUTOCOLLIMATION OF THE +X FACE OF EACH OPTICAL CUBE THROUGH THE RELAY MIRRORS AND THEIR AZIMUTH, ELEVATION, TABLE ROLL, AND DIHEDRAL MIRROR AUTOCOLLIMATION AZIMUTH ANGLES.

5. INVERT GRAVITY INVERSION FRAME AND REPEAT THE ABOVE MEASUREMENTS, BUT USE LOWER RELAY MIRRORS. THE +X CUBE FACES NOW BECOME THE -X CUBE FACES.
6. THESE RECORDED DATA ARE INPUT TO THE OAF COMPUTER PROGRAM, WHICH BY OPTICAL MATRICIES, AND DIRECTION COSINE OPERATIONS, PRODUCE THE ANGLES OF THE OPTICAL CUBE FACE NORMALS IN A SYSTEM OF CARTESIAN COORDINATE PLANES FOR ROLL PITCH AND YAW.

7. THE LISTING FROM THESE DATA RUNS, WHEN AVERAGED, GIVE THE "0 G" POINTING NORMALS FOR ALL NON-ALIGNED OPTICAL CUBES. COMPUTATIONS ARE NOW PERFORMED TO SIZE THE THICKNESS OF SEAT SHIMS AND DETERMINE THE "1 G" POINTING NORMALS TO POSITION EACH OPTICAL CUBE AND ITS ASSOCIATED INSTRUMENT INTO CO-ALIGNMENT WHEN FLANGE BOLTS ARE LOOSENED WITH THE INVERSION FRAME IN THE UPRIGHT POSITION. IT IS ASSUMED THAT OAF COORDINANTS CAN BE BACKED OUT OF THE COMPUTER PROGRAM SO THAT EACH OPTICAL CUBE CAN BE ADJUSTED TO ITS OWN OAF SET OF COORDINATES.
8. THE SHIMS ARE INSTALLED, ADJUSTMENTS ARE NOW PERFORMED USING MICROPOSITIONERS (THEODOLITE SET FOR "1 G" POINTING NORMALS), AND FLANGE BOLTS RETIGHTENED ON INDIVIDUAL EXPERIMENTS, EACH HAVING ITS OWN SET OF OAF COORDINATES.
9. THE SYSTEM ONCE AGAIN IS MEASURED AS IN STEPS 4 THROUGH 8, ITERATING AS NECESSARY TO ACHIEVE PROPER ALIGNMENT.
10. ALIGNMENT HAVING BEEN COMPLETED, MICROPOSITIONERS CAN BE REMOVED AND DOWELING OF ALL MOUNTING FLANGES CAN PROCEED.

11. THE GRAVITY INVERSION FRAME IS NOW PLACED IN THE UPRIGHT POSITION. THE CUBE FACE NORMAL ANGLES ARE ONCE AGAIN MEASURED IN THE OAF TO SERVE AS GAGING IN DETERMINING ALIGNMENT STABILITY, AND ALIGNMENT MAINTENANCE THROUGH ENVIRONMENTAL TESTING. OPTICAL MEASUREMENTS BY THE OAF AT THE COMPLETION OF THE TESTING PROGRAM WHEN COMPARED TO THE GAGE ALIGNMENT WILL DEMONSTRATE ALIGNMENT STABILITY.

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PRELIMINARY
INSTRUMENT COALIGNMENT ERROR BUDGET (ARCSEC)

<u>FIXED ERRORS</u>	<u>INTERFACE SPECIFICATION</u>	<u>TEMPLATE ALIGNMENT</u>	<u>PROPOSED ALIGNMENT</u>
MOUNTING ERROR	±10 (TEST)	? (ARCMIN)	±10
EFFECTS OF GRAVITY	±5 (ANALYSIS)	±5	±5
THERMAL MISALIGNMENT	±15 (ANALYSIS)	±5	±5
MEASUREMENT/INSTRUMENTATION ERROR	±5 (ANALYSIS)	±5 x ?	±5 x 2
EFFECTS OF VIBRATION (PITCH OR YAW)	±35 (TEST)		
EFFECTS OF VIBRATION (ROLL)	±10 (TEST)		
<u>STABILITY (PER 5 MIN. OF TIME)</u>			
THERMAL STABILITY	±1.0	±5.0	±5.0
MECHANICAL STABILITY	±0.5		
ELECTRICAL STABILITY	±0		

14

● DESIGN LOADS

- STATIC LOADS BASED ON S-320-G AND JSC 07700
- DYNAMIC INPUT ASSUMMED TO BE 4.56G

● MARGINS OF SAFETY

- BASED ON MICROYIELD CRITERION
- NO FACTOR OF SAFETY USED

● INSTRUMENT BOX STIFFNESS

- BOXES ARE MODELED WITH 0.125" QUAD-PLATES
- BOX STIFFNESS AFFECTS PANEL LOADS
- FIXED MOUNT LOCATED AT "INBOARD-AFT" CORNER OF BOX

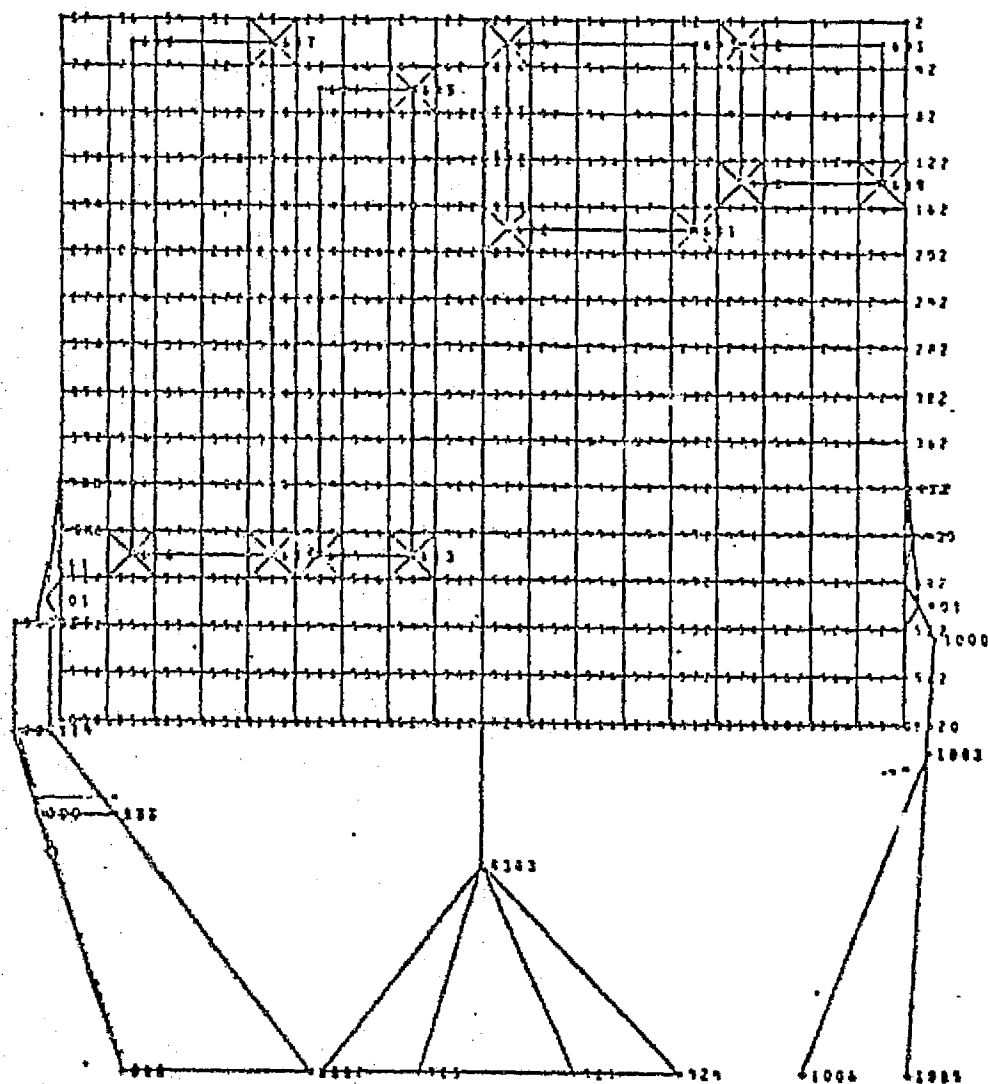
● FLEX MOUNT ANALYSIS

- EQUIVALENT BEAM MODEL
- LOCAL STRESSES NOT AVAILABLE.

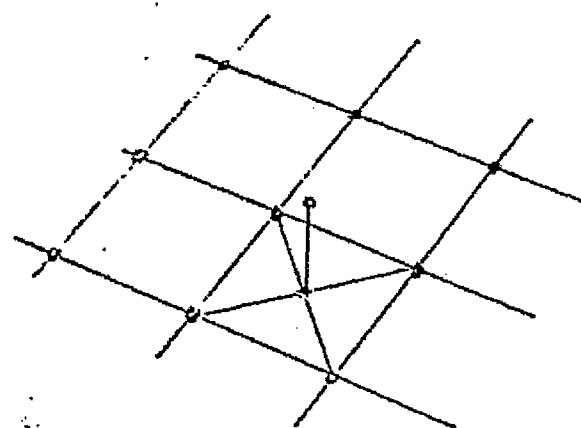
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RCA FINDINGS OF MEGA ANALYSIS



INSTRUMENT SUPPORT PANEL



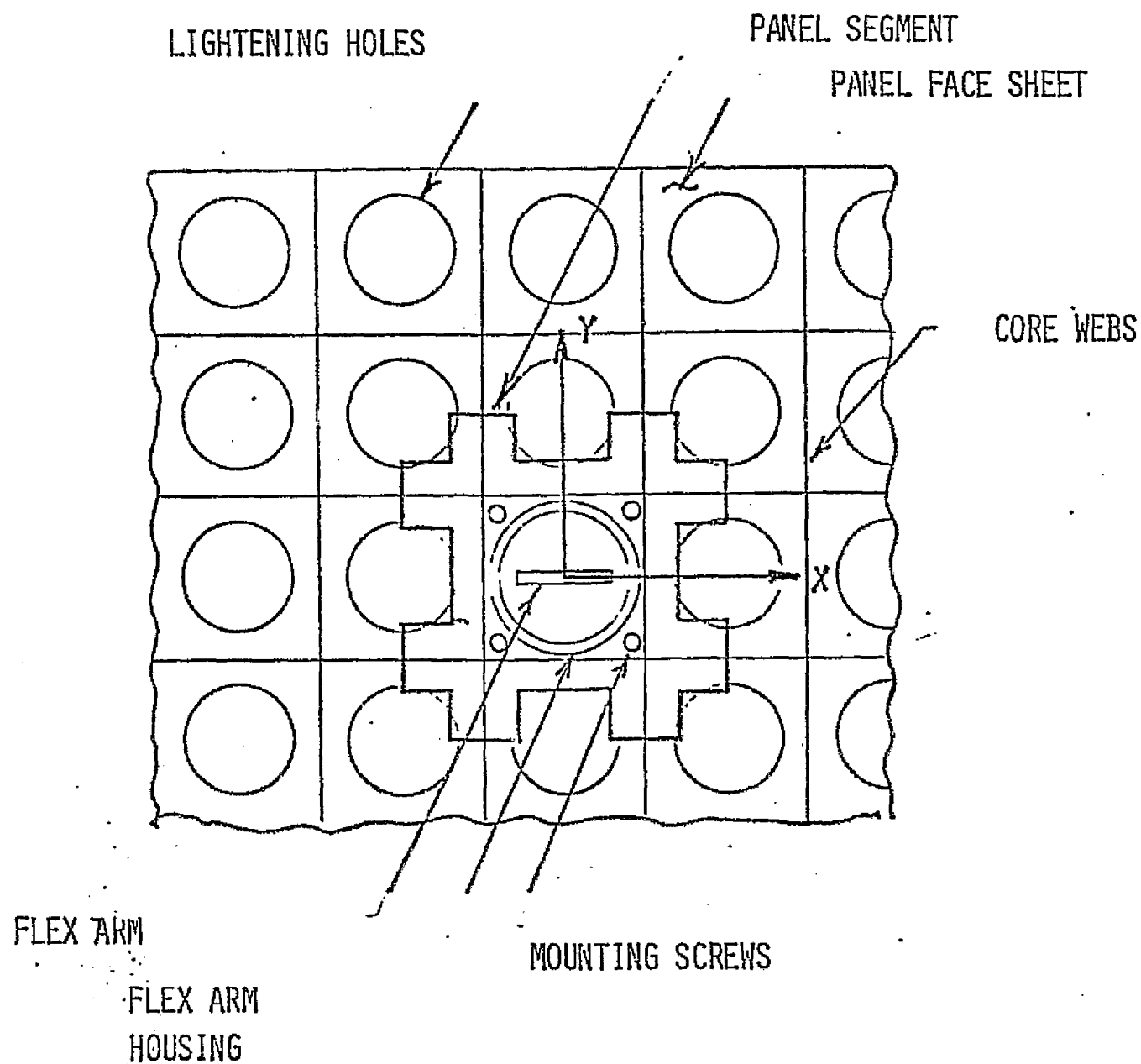
HOUSING

FLEX ARM

FLEX MOUNT DETAIL

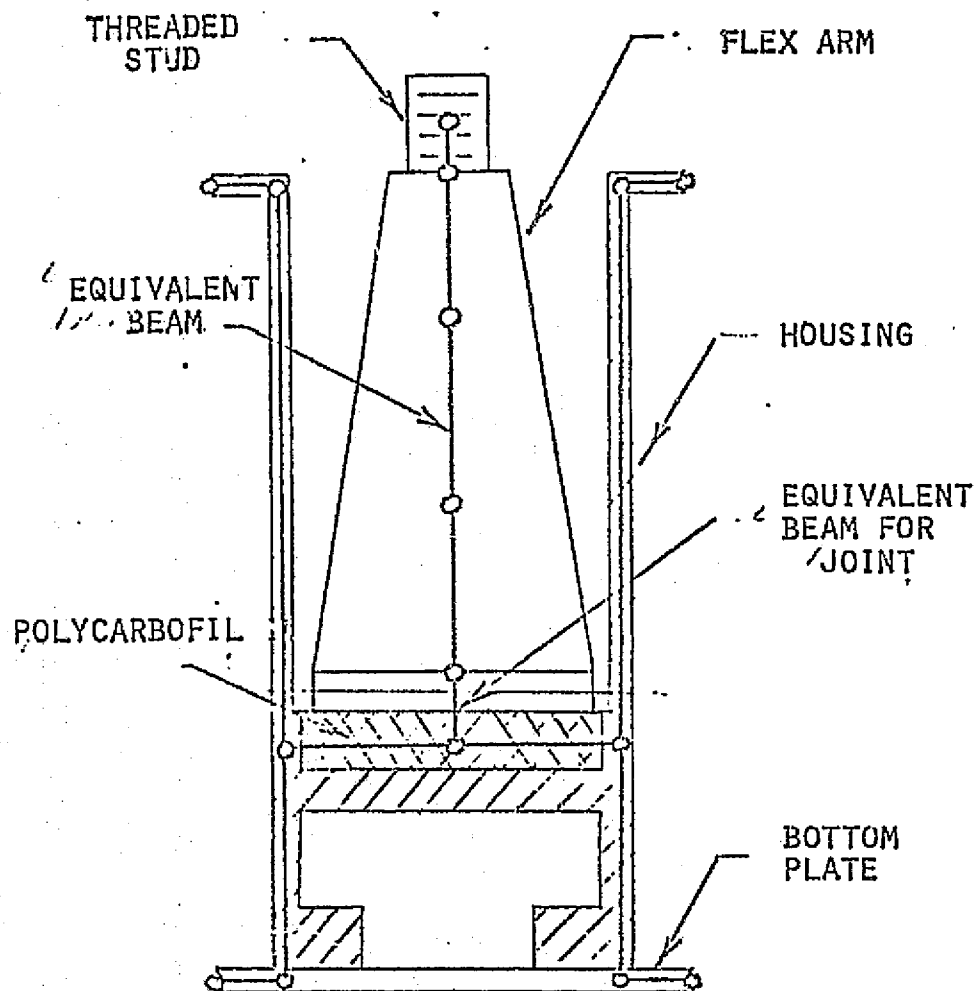
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PANEL SEGMENT
USED IN
FLEX MOUNT ANALYSIS

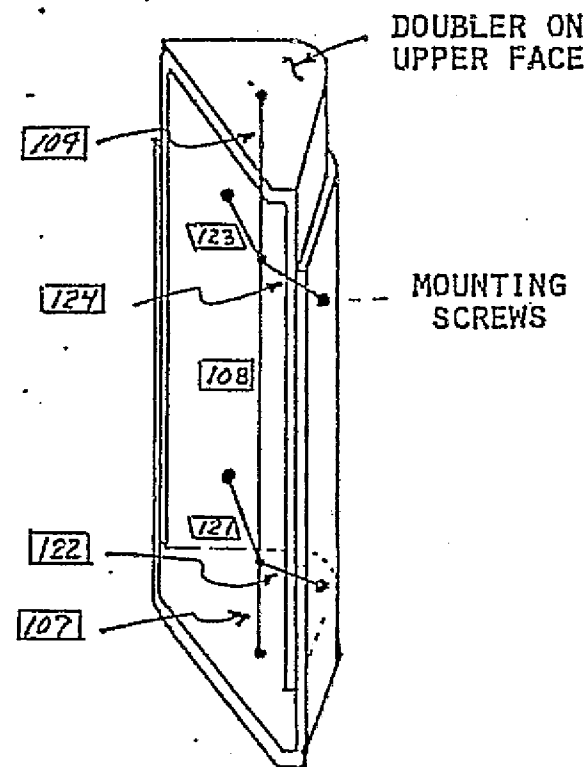


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EQUIVALENT BEAM MODELS OF COMPONENTS



FLEX ARM HOUSING

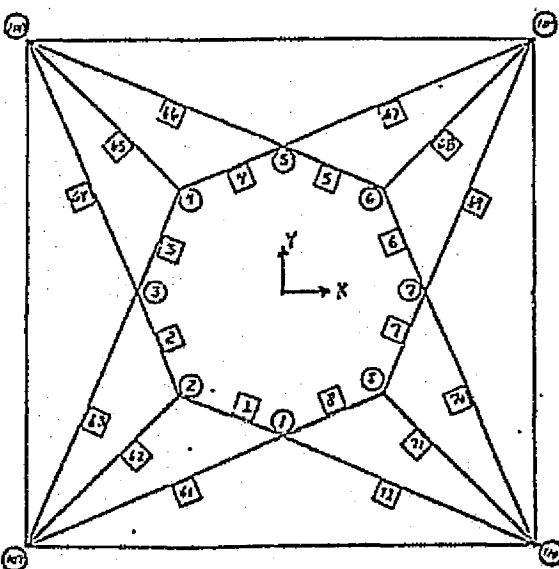


CORNER BRACKET

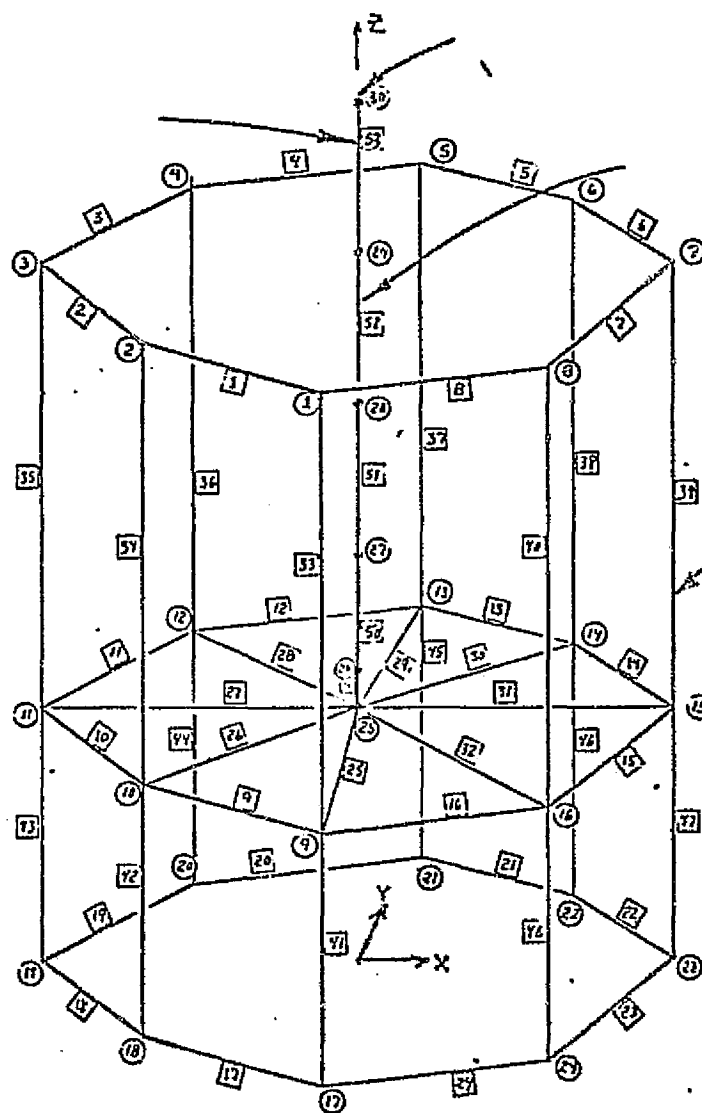
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FLEX ARM HOUSING FINITE ELEMENT MODEL

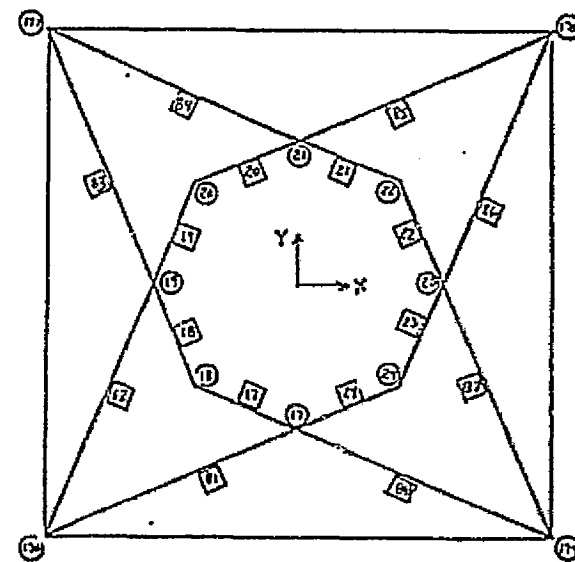
THREADED STUD



TOP PLATE



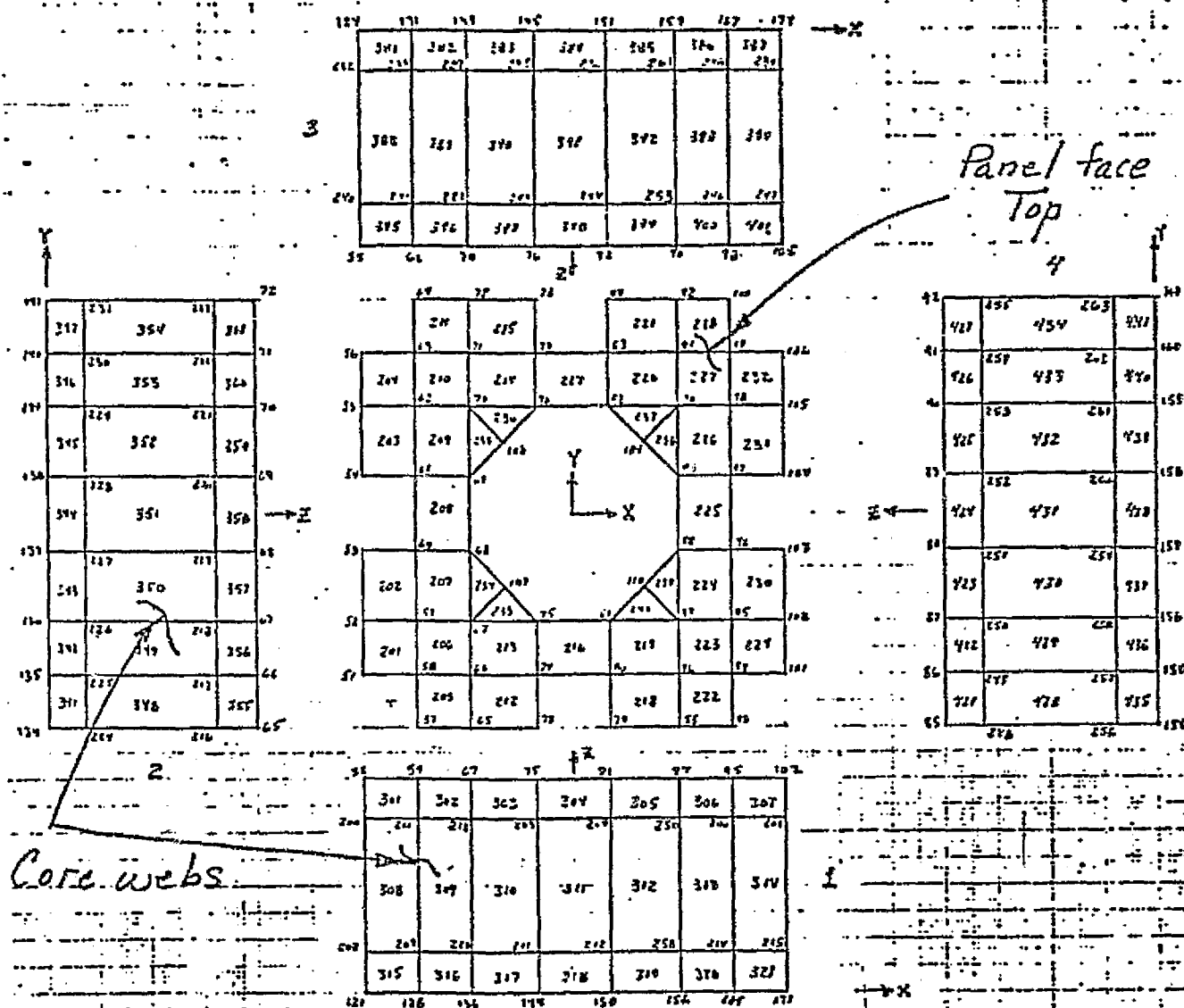
FLEX ARM



BOTTOM PLATE

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FINITE ELEMENT MODEL OF PANEL SEGMENT



LOWER FACE SHEET NOT SHOWN.

● STIFFNESS FOR THERMAL GROWTH:

• 0.020" DEFLECTION AT FLEX ARM PRODUCES:

-- 141. LB RESTRAINT LOAD ON INSTRUMENT

-- 374. PSI MAXIMUM STRESS IN TOP FACE PLATE

● LOADS AND STRESSES IN FOLLOWING MEMBERS

• FLEX ARM

• FLEX ARM MOUNTING SCREWS AND ALIGNMENT PINS

• FLEX ARM HOUSING

• CORNER BRACKETS AND MOUNTING SCREWS

• PANEL MOUNTING SCREWS AND ALIGNMENT PINS

• PANEL CORE AND FACE SHEETS

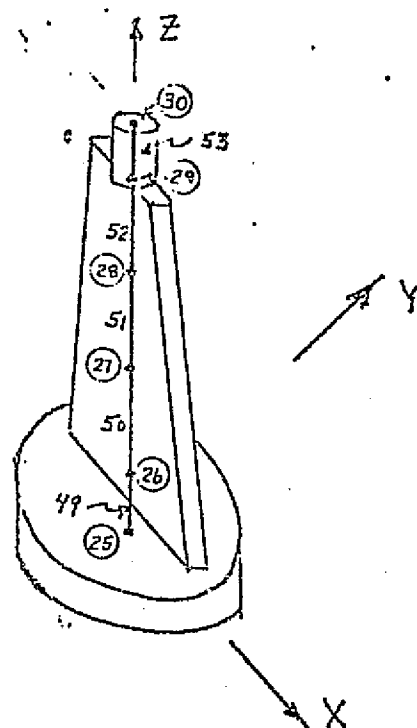
FINITE ELEMENT ANALYSIS

UNIT LOAD IN THREE
ORTHOGONAL AXES:

$$P_X = 1. \text{ LB}$$

$$P_Y = 1.$$

$$P_Z = 1.$$



DETAILED STRESS ANALYSIS

ONE LOAD CONDITION
DEFINED:

$$P_X = 2500. \text{ LB}$$

STRESS ANALYSIS SUMMARY
2500 LB LOAD ON FLEX ARM

NO.	LOCATION	LOADING	MARGIN OF SAFETY
1	MOUNTING STUD	SHEAR	+.74
2	BALL SEAT	BEARING	+.27
3	MOUNTING SCREW	BEARING	+.79
4	FLEX ARM	BENDING	+.31
5	FLEX ARM SCREWS	TENSION	+.10
6	FLEX ARM DOWELS	BENDING AND SHEAR	+.13
7	FLEX ARM DOWELS	BEARING	-.38 (+.69) ⁽¹⁾
8	HOUSING	BENDING	-.16 (+.22) ⁽²⁾
9	HOUSING SCREWS	TENSION	AMPLE
10	HOUSING DOWELS	BEARING	-.92 (AMPLE) ⁽¹⁾
11	FACE SHEETS	BENDING	+.89
12	CORNER BRACKET SCREWS	TENSION	AMPLE

(1) INCLUDES FRICTION OF ONE ADJACENT SCREW.

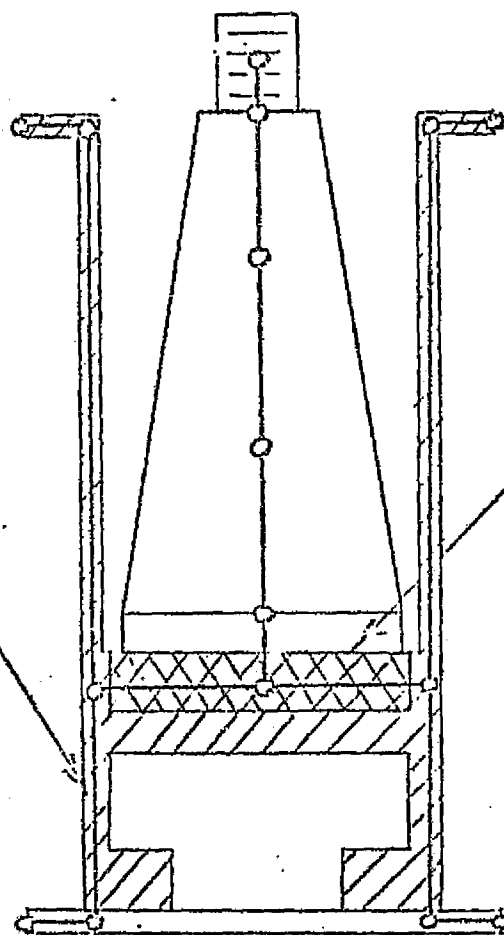
(2) REQUIRES A 0.005 IN. INCREASE IN HOUSING WALL THICKNESS.

BENDING IN HOUSING

MS = -.16

CORRECTION:

INCREASE WALL
WALL THICKNESS BY
0.005 IN.



FLEX ARM DOWELS:

CONTACT STRESSES IN
ALUMINUM BASE

MS = -.38

CORRECTION:

FRICTION OF ONE
ADJACENT SCREW

MS = +.69

BASIC DESIGN CRITERIA

- PUT THE ADJUSTMENT REQUIREMENT PROVISION ON EACH EXPERIMENT.
- TWO TYPES OF ADJUSTMENT USED - BOTH COMPLEX - MICROTHREAD PIVOT TYPE.
- INTERFACE TEMPLATES ADOPTED - WITH DOWEL REGISTRATION.
 - DESIGNED TO CONTROL EACH EXPERIMENT TO CRUCIFORM TO WITHIN ± 20 ARC.MINS.
 - EXPERIMENT ADJUSTMENT CAPABILITY DESIGNED TO COVER THE SAME ERROR
- OPTICAL ALIGNMENT CUBES ON EACH EXPERIMENT - SOME EXPERIMENTS HAD BORESIGHT ADJUSTMENT TO THEIR OWN CUBE.
- USED ONE STANDARD BOLT SIZE - MOUNTING FEET TO CRUCIFORM.
 - SNUG FIT ON BOLTS - USED STANDARD TORQUE NUTS.
- GOAL - ASSEMBLE, ALIGN AND FLY AT 68°F .
- REQUIREMENT ON NRL-B AND HCO-A TO FSS 10 ARC.SECONDS.

ALL THREE WERE MOUNTED ON SOLID SPAR OF CRUCIFORM.

ALIGNMENT METHOD

- SUSPENDED TOTAL PACKAGE FROM A TOP FIXTURE - SUN VIEW END DOWN .
- EACH EXPERIMENT CRADLED WITH SLING ON CENTERLINE OF CENTRE OF MASS
INCLUDING A LOAD CELL.
- ASSEMBLED FSS TO CRUCIFORM.
 - ASSEMBLED EACH EXPERIMENT - OFFLOADING IT'S MASS AT THE LOAD CELL.
 - ALIGNED EACH EXPERIMENT - ADJUSTMENT OR SHIMMING - '0' G POSITION.
UNTIL ALL CO-ALIGNED TO FSS TO WITHIN ALIGNMENT TOLERANCE.
 - RELEASED ALL LOAD CELLS.
 - TOOK A COMPLETE SET OF READINGS. THIS BECAME 1G BASELINE USED FOR
ALIGNMENT CHECK BEFORE AND AFTER ENVIRONMENTAL TESTING.
 - WENT THROUGH VIB AND THERMAL VACUUM AT HOUSTON.
 - POST ENVIRONMENT CHECK AT HUNTSVILLE.
 - CLAIM ONLY MINOR ADJUSTMENT NECESSARY.
 - NO ADJUSTMENT AT THE CAPE.



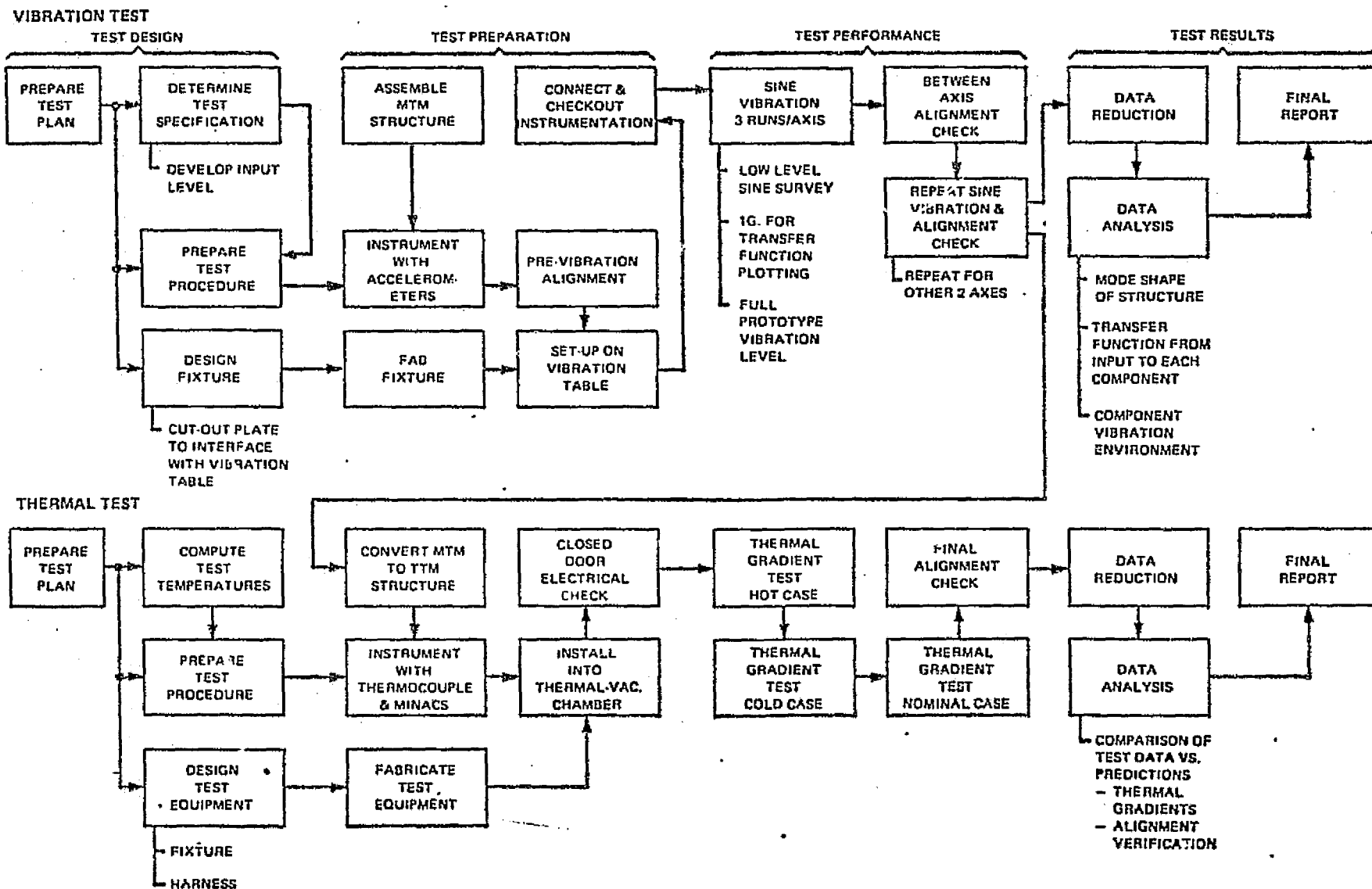
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A.T.M. INVESTIGATION

45

ALIGNMENT PROCEDURE

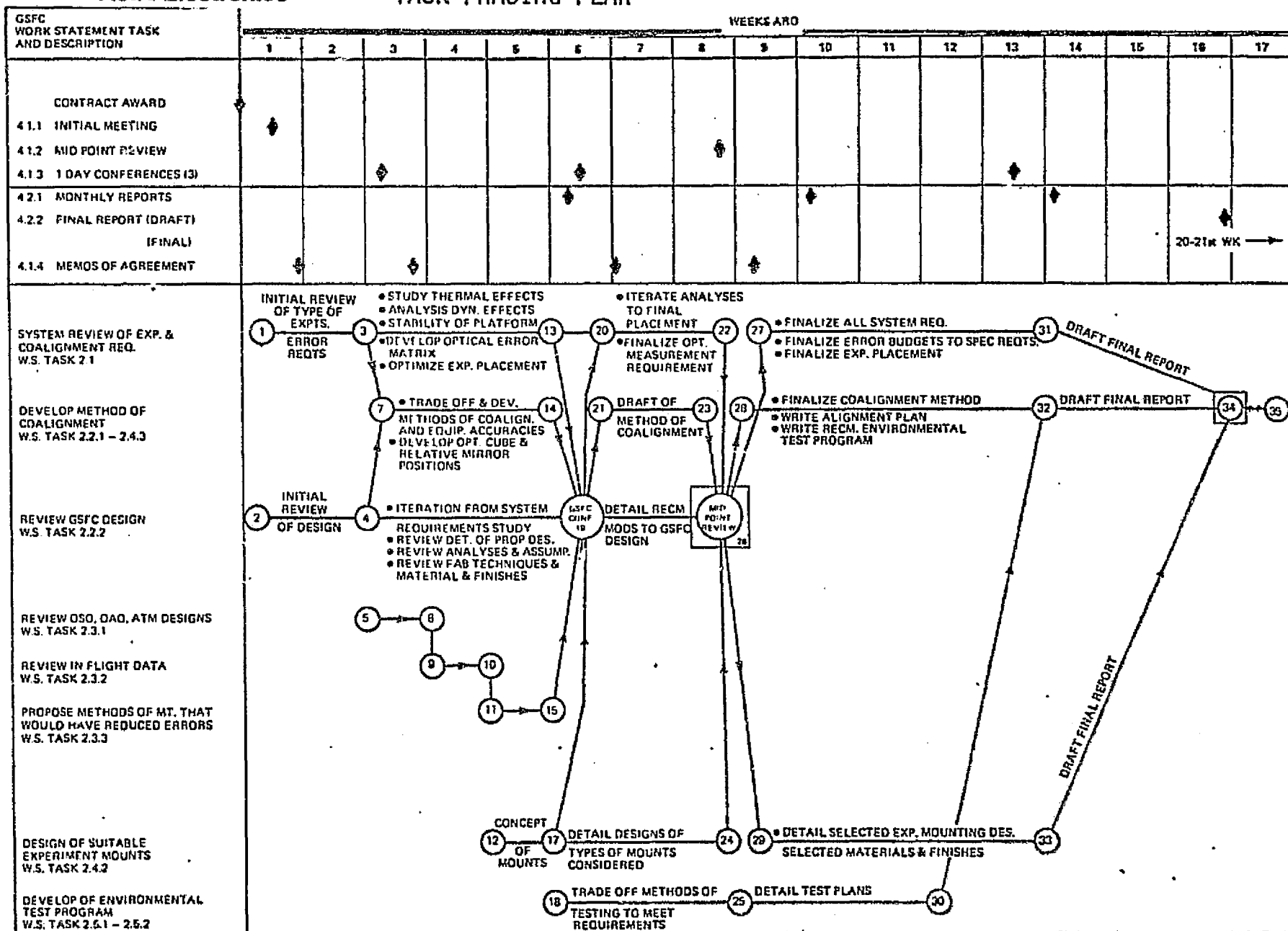
- VERY LENGTHY PROCEDURE
 - ALLOWED 24 HR. DWELL TIME FOR EACH SET OF EXPERIMENT OPTICAL MEASUREMENTS.
 - WENT ROUND THE EXPERIMENT MEASUREMENT AND ADJUSTMENT LOOP THREE TIMES AT LEAST.
- CLAIM -- APPROX. 10 DAYS FOR INITIAL ALIGNMENT AFTER ALL EXPERIMENTS WERE INSTALLED.
- CLAIM -- ALIGNMENT CHECKS THEREAFTER 8 HOURS ONCE THEY WERE SET UP.
- FINAL COMMENTS
 - FELT VERY STRONGLY THAT EACH P.I. SHOULD BE RESPONSIBLE FOR ADJUSTMENT TO FINAL ACCURACY.
 - DID TAKE EXPERIMENTS OFF, RE-BORESIGHT AND REPLACE WITH NO PROBLEMS.
 - DID OBSERVE SOME MOVEMENT -- POWER 'ON' TO POWER 'OFF'.
 - STATED THAT ALIGNMENTS SHIFTS 'ON ORBIT' HELD TO BETTER THAN 3 ARC.SECONDS.





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TASK PHASING PLAN



REQUIREMENTS TO COMPLETE STUDY

48

- FINAL EXPERIMENT LOADING CRITERIA
- NASA SELECTION OF MOUNT DESIGN
- NASA THERMAL TRANSIENT DATA
- CONFIRMATION OF CONFIGURATION 'A' AND
FINAL POSITION OF FIXED AND FLEX MOUNTS.

APPENDIX G

DETAIL AND ASSEMBLY DRAWING
MODIFICATION RECOMMENDATIONS



To R. Packer Location MS111 Date 4 February 1976

From F. Gross Location MS111 Telephone 3496

Subject SMM - Flex & Fixed Mount Drawing Revisions

Prints of the latest flex and fixed mount designs have been reviewed with regard to recommendations made at the mid-point review and new recommendations resulting from the recent loads and stress analyses. The following list of drawing changes provides a single flex and a single fixed mount design optimized for the maximum load condition and satisfying the agreed to margin of safety requirements at all mounts. If weight and thermal conductivity prevail against commonality, a second set of mounts for the lighter instruments could be added to the design by tabulating blade thickness only. In the absence of assembly drawings a listing has been added to include the recommendations associated with the assemblies. R. Federline indicated that these assembly drawings would be produced by GSFC in the near future. In addition, lower tier assembly drawings are recommended to match the spherical seats to the flex and fixed member pilots to provide the minimized clearances required by the alignment budget.

The GSFC approach to producing bearings and seats with known eccentricities to correct for errors in the final alignment is not compatible with the requirement to match the spherical seat with the mount pilot. The clearances associated with these interchangeable parts are not compatible with the alignment budget. The following recommendations delete these parts with known eccentricities. A possible alternate approach would be to produce eccentric replacement seats only as required in the final alignment and lap them to match the existing previously installed pilots.

GD 1085215 Spherical Bearing Flex Mount

1. Delete note 6 and reference on pictorial.
2. Delete ϕ of B and ϕ of C scribe line view.
3. Delete tabulation.

To: R. Packer

4 February 1976

4. Change A to .438
 " B to .5630 $\begin{matrix} +.0000 \\ -.0005 \end{matrix}$ (Retain B designation)
 " C to .4375 $\begin{matrix} +.0005 \\ -.0005 \end{matrix}$ (Retain C designation)
 " D to .219
5. Add to Surface B $\begin{matrix} \text{C} \\ \text{C} \end{matrix} \begin{matrix} .0001 \\ .005 \end{matrix}$
 -1 -2
6. Add to C Designation, "Spherical Radius $\begin{matrix} \triangle \\ .0001 \end{matrix}$ "
7. Shorten Note 7 to "Scribe dash no. in approximate position shown."
8. Delete -XX in pictorial.
9. Add mark to identify dash no. after installation in instrument flange.

GD 1085216 Spherical Seat Flex Mount

- Delete "on radius of maximum eccentricity" from Note 6.
Add "on Dash 1 only".
- Delete "followed by amount of eccentricity in ten-thousandths if less than .0001 scribe 01 after dash number" from note 7.
- Delete ϕ of C and ϕ of A scribe line view.
- Delete existing tabulation.
- Delete -XX in part marking view.
- Tabulate dimension A only for two dash numbers. -1 for lower seat which will be matched to flex member at next assembly. -2 for upper seat.

DASH NO.	A DIA.	$\begin{matrix} \text{C} \\ \text{A} \\ \text{C} \end{matrix}$
-1	$\begin{matrix} +.0005 \\ .4368 \end{matrix}$ $\begin{matrix} -.0000 \\ \end{matrix}$	.0001
-2	.438 $\begin{matrix} +.005 \\ \end{matrix}$	.005

R. Packer

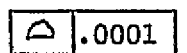
4 February 1976

GD1085216 (continued)

- 7. Change B to .875
- " C to $.4375 \pm .0005$ (Retain C Designation)
- " D to .312
- " E to .562

8. Change Title to "Spherical Seat Flex and Fixed Mount"

9. Add to C Designation, "Spherical Radius



"

GF 1085086 Light Flex Member SMM

1. Delete tabulation

2. Change A to 4.635

" B to .875 Dia.

" C to $.4380 \begin{matrix} +.0000 \\ -.0005 \end{matrix}$ Dia.

" D to .375 Dia

" E to .375-24 UNF-2A

" F to TBD

" G to .281R

" H to 3.105

3. Delete "Light" from Drawing Title.

4. Show undercut in side view.

5. Add witness scribe line on B Dia Vertical surface (one location)

6. Add 4 microinch surface finish to C Diameter.

7. Add .594 Dia Spotface (Backface) to four holes.

8. Change mounting hole locations:

R. Packer

4 February 1976

8. (con't)

From .625 to .500

From 1.250 to 1.000

From .437 to .500

From .875 to 1.000

GF-1085026 Fixed Member

1. Change TBD overall height to 4.635.
2. Show hidden cruciform intersection with .125 radii in top view.
3. Add two .031 Dia dowel pin vent holes .20 deep radially on 2.000 dia located .235 up from base. Holes lie on vertical ϕ in front and top views.
4. Add witness scribe line on .875 dia. vertical surface (one location).
5. Delete 8 microinch surface finish (2 views).
6. Add 4 microinch surface finish to .4380 dia.
7. Add .594 Dia spotface (Backface) to four holes.
8. Change blade thickness to TBD.
9. Change mounting hole locations:

From .625 to .500

From 1.250 to 1.000

From .437 to .500

From .875 to 1.000

h-10

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GC 1085203 Thermal Insulator Light Flex Mount SMM

1. Change title to "Mount Thermal Insulator SMM".
2. Change Polycarbofil to Polycarbafil.
3. Add .030 x 45° chamfers to 2.093 dia. Revise Note 3 to suit.
4. Change hole locations:
From 1.250 to 1.000
From .875 to 1.000

GD 1085201 Housing, Light Flex Mount

1. Change title to "Housing, Mount".
+.004
2. Change .055 -.000 Pilot Holes to .093.
3. Change .031 $\begin{smallmatrix} +.004 \\ -.000 \end{smallmatrix}$ Pilot Holes to .250.
4. Delete 8 microinch surface finish in bore.
5. Change hole locations for Item 2:
From .625 to .500
From 1.250 to 1.000
From .437 to .500
From .875 to 1.000

GC 1085068 Bottom Plate Experiment Mount

1. Change .055 $\begin{smallmatrix} +.004 \\ -.000 \end{smallmatrix}$ Pilot Holes to .093.
+.004
2. Change .025 -.000 Pilot Holes to .093.

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GC1085024 Shim, Light Flex Member

1. Change title to "Shim, Mount"
2. Delete tabulation
3. Change A to .875
B to .500
4. Add to Note 2 ".032 thick"
5. Change thickness on pictorial to ".032(REF.)"

GC105023 Shim, Bottom Plate

1. Add 2.375 R to match bottom plate corners.
2. Change 2.500 dia to 2.530 dia.
3. Add to note 2 ".062 thick".
4. Change thickness on pictorial to ".062(Ref.)"

TBS Assembly, Flex Member/Seat

1. Lap bore of spherical seat GD1085216-1 to provide a clearance fit of 50 to 100 microinches on the diameter when seated on flex member GF1085086.
2. Serialize as matched set.
3. Lubricate mating surfaces lightly with SRG-60 oil or equivalent.

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TBS Assembly, Fixed Member/Seat

1. Lap Bore of spherical seat GD1085216-1 to provide a clearance fit of 50 to 100 microinches on the diameter when seated on fixed member GF 1085026.
2. Serialize as matched set.
3. Lubricate mating surfaces lightly with SRG-60 oil or equivalent.

TBS Assembly, Flex Mount

1. Fasten Flex member to housing with four .312-24 x .938 long titanium screws (HIT1205V6).375 Grip.
2. Torque titanium screws to TBD in-lb.
3. Dowel flex member to housing with two 5/16 dia. dowels, 1.00 long (MS16555-653).
4. Provide 3/8*24 locknut (NAS-1291C6M) on flex member stud.
5. Include mount shim and upper spherical seat in assembly.
6. "Stake" dowels with structural adhesive (3M 1838 or equivalent) tacks.

TBS Assembly, Fixed Mount

1. Flat bottom drill and ream .980 deep for dowel pins.
2. Dowel Fixed member to housing with two 5/16 dia. dowels, 1.00 long (MS16555-653).
3. All other assembly considerations to be the same as the flex mount assembly.

Mr. Packer

4 February 1976

TBS Installation, Fixed & Flex Mounts

1. Dowel bottom plate to housing with four 1/8 dia dowels, .375 long (MS16555-625).
2. Dowel bottom plate and housing to platform with four (each) 1/8 dia. dowels .375 long (MS16555-625).
3. Size bottom plate shim at assembly.
4. Key final installation and doweling to alignment procedure.
5. Torque titanium (HIT1203V8) mounting screws to TBD in-lb.
6. "Stake" dowels with structural adhesive (3M1838 or equivalent) tacks.
7. Provide note to align witness marks on spherical seats with marks on mating fixed and flex members.
8. Torque 3/8-24 locknut to TBD in-lb. above locking feature torque.

F. Gross

F. Gross

FG:gp

APPENDIX H

TEMPLATE ALIGNMENT TOLERANCES

TEMPLATE ALIGNMENT TOLERANCES

(ABOUT AXIS NORMAL TO INSTRUMENT MOUNTING PLANE)

MASTER & INDIVIDUAL TEMPLATE CONCEPT

OPERATION, TRANSFER OR FIT				INSTRUMENT			FSS		
				DIAMETRAL TOLERANCE	SPAN	ERROR	DIAMETRAL TOLERANCE	SPAN	ERROR
				(IN)	(IN)	(ARCSEC)	(IN)	(IN)	(ARCSEC)
1 DOWEL MASTER TEMPLATE TO ISP				.0008	92	± 1.8	—	—	—
2 DOWEL INDIVIDUAL TEMPLATE TO MASTER TEMPLATE				.0008	12	± 13.8	.0008	12	± 13.8
3 INSTALL CUBE ON MASTER TEMPLATE				—	—	—	—	—	—
4 INSTALL CUBE ON INDIVIDUAL TEMPLATE				—	—	—	—	—	—
5 DRILL INDIVIDUAL TEMPLATE				—	—	—	—	—	—
6 BUSH INDIVIDUAL TEMPLATE				—	—	—	—	—	—
7 MOUNT INDIVIDUAL TEMPLATE CUBE TO MASTER TEMPLATE CUBE				—	—	± 2	—	—	± 2
8 DRILL INSTRUMENT FLANGE				.0005	12	—	.0005	8	—
9 PRESS SPHERICAL BEARINGS IN INSTRUMENT FLANGE				—	—	—	—	—	—
10 @ SPHERICAL SURFACE TO PRESS FIT PILOT				.0002	12	—	.0002	8	—
SLEEVE ADAPTER TO BUSHING				.0007	12	± 12.0	.0007	8	± 18.0
12 SLEEVE ADAPTER @ O.D. TO I.D.				.0002	12	± 3.4	.0002	8	± 5.2
SLEEVE ADAPTER TO MOUNT PILOT				.0004	12	± 6.9	.0004	8	± 10.3
14 DOWEL MOUNT BUSHING TO ISP				—	—	—	—	—	—
STUD ADAPTER TO BEARINGS SEAT				.0004	12	± 6.9	.0004	8	± 10.3
@ STUD ADAPTER PILOTS				.0002	12	± 3.4	.0002	8	± 5.2
17 STUD ADAPTER TO BUSHING				.0012	12	± 20.6	.0012	8	± 30.9
ALIGN BORESIGHT TO INDIVIDUAL TEMPLATE CUBE				—	—	$\pm 2^*$	—	—	$\pm 2^*$
19 BEARING SEAT BORE TO MOUNT PILOT				.0001	12	± 1.7	.0001	8	± 2.6
@ BEARING SEAT BORE TO BEARING SEAT				.0002	12	—	.0002	8	—
TOTAL ERROR						± 74.5			± 100.3
TOTAL ERROR = $\pm 174.8 \text{ SEC}$ = $\pm 2.91 \text{ MIN}$									
TOTAL RSS ERROR = $\pm 51.4 \text{ SEC}$									
* ASSUMES BORESIGHT IS DIRECTLY ALIGNABLE									

BASIS FOR DIAMETRAL TOLERANCES:

- 1- .001 OVERSIZE REAMER, .0002 OVERSIZE DOWELS
- 2- .001 OVERSIZE REAMER, .0002 OVERSIZE DOWELS
- 8- DRILL BUSHING $+0.0001$ TO $+0.0005$
- 10- SPHERICAL SURFACE $\odot$ TO ALLOW FIT PILOT WITHIN .0002
- 11- DRILL BUSHING $+0.0001$ TO $+0.0005$, SLEEVE O.D. .0000 TO -0.0002
- 12- O.D. $\odot$ TO I.D. WITHIN .0002
- 13- SLEEVE I.D. $+0.0001$ TO $+0.0003$, MOUNT PILOT .0000 TO -0.0001
- 15- STUD -0.0001 TO -0.0003 , BEARING SEAT BORE .0000 TO $+0.0001$
- 16- BUSHING PILOT $\odot$ TO BEARING SEAT PILOT WITHIN .0002
- 17- BUSHING $+0.0001$ TO $+0.0005$, STUD -0.0005 TO -0.0007

THIS IS TO ASSURE ENGAGING BOTH PILOTS FOR WORST CASE TOLERANCES OF DRILLED HOLE, BEARING, AND BEARING SEAT.

- 19- .0001 DIAMETRAL CLEARANCE BETWEEN BEARING SEAT BORE AND MOUNT PILOT (FABRICATED AS MATCHED SET.)

IN 13/7

Astro Electronics
Division
Princeton, New Jersey

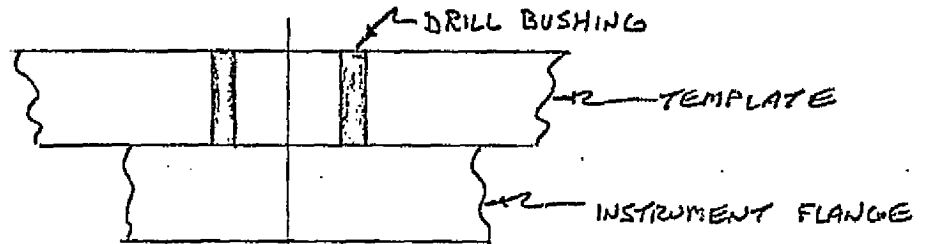
Prepared for: _____ Report No. _____

Approved: _____ Project: _____

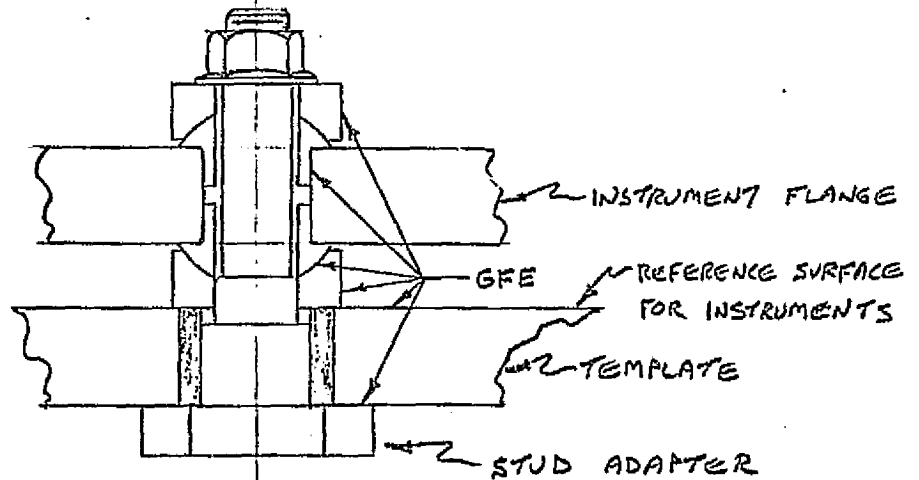
Date: _____ Contract No. _____

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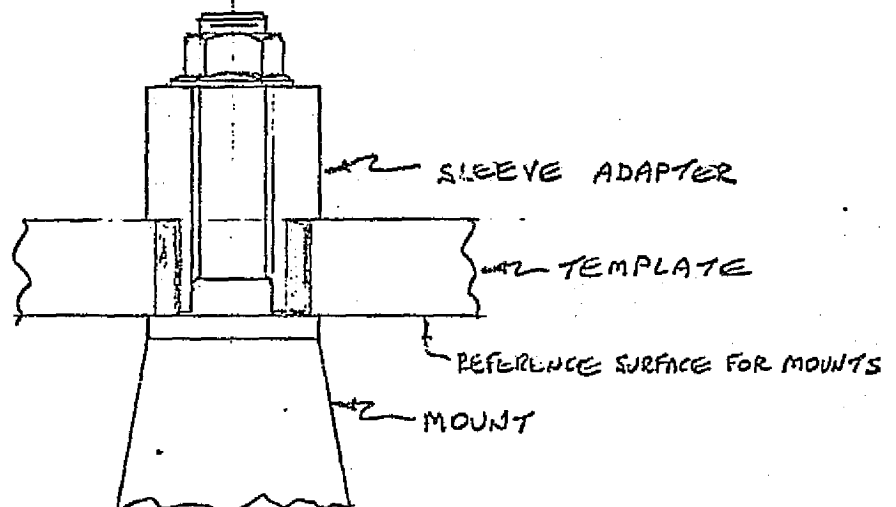
DRILL INSTRUMENT
FLANGE



ALIGN BORESIGHT
TO TEMPLATE
CUBE (AT P.I.'s
FACILITY)



INSTALL MOUNTS
IN ISP



TEMPLATE ALIGNMENT TOLERANCES

(ABOUT AXIS NORMAL TO INSTRUMENT MOUNTING PLANE)

DUAL INDIVIDUAL TEMPLATE CONCEPT

OPERATION, TRANSFER OR FIT	INSTRUMENT			FSS		
	DIAMETRAL TOLERANCE	SPAN	ERROR	DIAMETRAL TOLERANCE	SPAN	ERROR
	(IN)	(IN)	(ARCSEC)	(IN)	(IN)	(ARCSEC)
DOWEL DUAL TEMPLATE BLANKS TOGETHER	—	—	—	—	—	—
DRILL & JIG GRIND TEMPLATES TOGETHER	—	—	—	—	—	—
PRESS SPECIAL SPHERICAL BEARINGS IN NASA TEMPLATE	—	—	—	—	—	—
PRESS SPECIAL SPHERICAL SEAT IN GFE TEMPLATE	—	—	—	—	—	—
ASSEMBLE TEMPLATES & INSTALL CURVE ONE EACH	—	—	—	—	—	—
MEASURE GFE TEMPLATE CURVE TO NASA TEMPLATE CURVE	—	—	± 2	—	—	± 2
JIG GRIND INSTRUMENT FLANGE FROM GFE TEMPLATE	.0001	12	—	.0001	8	—
PRESS SPHERICAL BEARINGS IN INSTRUMENT FLANGE	—	—	—	—	—	—
MEASURE BORESIGHT TO TEMPLATE CURVE	—	—	$\pm 2^*$	—	—	$\pm 2^*$
ALIGN INSTRUMENT TEMPLATE CURVE TO FSS TEMPLATE CURVE	—	—	$\pm$	—	—	± 2
INSTALL MOUNTS IN ISP (SEAT CLEARANCE)	.0001	12	± 1.7	.0001	8	± 2.6
DOWEL MOUNT HOUSINGS TO ISP	—	—	—	—	—	—
INSTALL INSTRUMENT (SEAT CLEARANCE)	.0001	12	± 1.7	.0001	8	± 2.6

TOTAL ERROR

± 9.4

± 11.2

TOTAL ERROR = ± 20.6 SEC

TOTAL RSS ERROR = ± 6.6 SEC

* ASSUMES BORESIGHT IS DIRECTLY MEASURABLE

RESIDUAL OFFSETS BETWEEN THE INSTRUMENT BORESIGHT AND THE GFE TEMPLATE CURVE WILL BE FURNISHED BY THE P.I. TO THE OPTICAL ALIGNMENT FACILITY. THIS OFFSET ALONG WITH THE MEASURED RESIDUAL OFFSET BETWEEN THE GFE TEMPLATE CURVE AND THE NASA TEMPLATE CURVE WILL BE USED TO ALIGN THE NASA INSTRUMENT TEMPLATE (TEMPLATE SYSTEM ALIGNMENT) TO THE NASA FSS TEMPLATE. THE CURVE ON THE NASA FSS TEMPLATE IS THE PRIMARY REFERENCE.

ROB.M

Astro Electronics
Division

Princeton, New Jersey

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Subject: SMM TEMPLATE ALIGNMENT Page No. 2 of 2

